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DIRECT-ACTING STEAM PUMPS

BY

FRANK F. NICKEL
ASSOCIATE IN MECHANICAL ENGINEERING,
COLUMBIA UNIVERSITY IN THE CITY OF NEW YORK

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PREFACE

The direct-acting pump has received scant attention from writers on pumping machinery. While this class of machinery in general is looked upon abroad with contempt and as a sort of necessary evil, in this country its advantages over other types of pumps were recognized early, even in the face of lower economy in fuel. The abundance of fuel and the scarcity of skilled labor, no doubt, were deciding factors in this development.

Within the past fifteen years the field has been encroached upon greatly by the centrifugal pump, but there are many conditions, especially where the exhaust steam is usable, in which the direct-acting pump is unlikely to be displaced. Evidence of this is found in many modern plants where direct-acting pumps are found side-by-side with centrifugal pumps, each supplying the service for which it is best fitted.

The author has considered it a duty to preserve the history of the development of this class of pump and the results of his experience, extending over thirty years in this line, and is confident that this book will supply much information that is looked for elsewhere in vain. In the few cases where material was used that is not original, due credit is given.

Grateful acknowledgment is made to Mr. Alfred D. Blake, Associate Editor, *Power*, for his valuable assistance in editing and for many excellent suggestions.

Frank F. Nickel.

East Orange, N. J., 1915.

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DIRECT-ACTING STEAM PUMPS

CHAPTER I

DEVELOPMENT OF THE DIRECT-ACTING PUMP

"In the early fifties of the last century," to quote F. W. Jenkins, "there was a little machine shop situated close to the East River on the line dividing Williamsburg from Brooklyn and opposite the old Navy Yard block; the name of the firm was Worthington & Baker. William H. Baker was a native of England and a fairly good mechanic. He had received little education in his youth, but was possessed of a vast fund of good judgment and common sense. Mr. Worthington was a bright and active young man with the power to rapidly concentrate on any business; expert in the use of tools and a good draughtsman."

He was the first man to realize the importance of a machine that would pump water without the intervention of a crankshaft or rotary motion. This idea appears very simple today, yet while he was endeavoring to solve this problem, he was, to use his own words, "nearly reduced to hopeless inaction by a remark of so good an authority as James D. Allaire, who said he considered it his duty to tell me that I was trying to invent a machine that was not wanted; that no part of the machinery, either for steamboats or for factory purposes, was more satisfactory than that for pumping. And this, at that time, appeared to be the opinion of every engineer."

Mr. Worthington's first efforts were directed toward producing a single, single-acting pump as the simplest form imaginable. The first pump built was intended for automatically feeding a boiler and is illustrated in Fig. 1. It was equipped with a spring-thrown valve.

1. Spring-thrown Valve.—Here was embodied the correct principle of a single pump valve motion which renders the action of the slide valve certain without depending upon the momentum of the engine. The momentum, especially when running slow, is frequently insufficient for this purpose, and with a valve rigidly connected to the piston, the latter will move just far enough to let the valve cover all the ports, when the pump will stop.

The steam supply to this pump was regulated by a float in the boiler, as shown in Fig. 2, and the pump was supposed to keep the water at a constant level. But, probably because the pump was single-acting, its motion was irregular and its action unreliable.

The steam cylinder was double-acting, rocker arm *a* (Fig. 1) operating the slide valve in the steam chest. This rocker arm was moved by a wristplate *b*, which imparted a motion opposite to that of the piston, this reversal of motion being essential when an ordinary slide valve of the "*D*" type is used. The wristplate had three pins, the two upper ones *c* moving the rocker arm and the lower one *d* being moved by two tappets *e* attached to the rod *f*, which in turn was guided and moved by an arm *g*, extending from the piston rod *h*. The piston, in traveling to the right, moved pin *d* to the right until at mid-position the valve was closed, and the steam supply cut off. In doing so, however, the pin *d* with its roller depressed the spring-loaded triangular cam *i* and, as the piston by virtue of its momentum would travel a little over the center, the roller, in sliding down the other side of the cam, would throw over the wristplate to the other side, opening the valve fully for the reversed stroke.

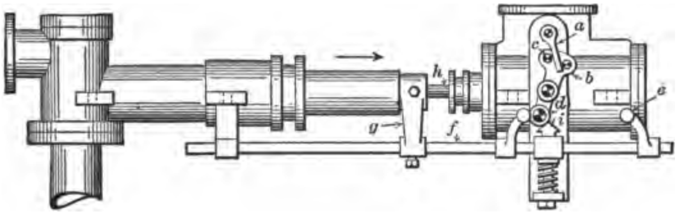


FIG. 1.—Single, single-acting feed pump with spring-thrown valve (Worthington, 1841).

The spring-loaded cam embodied the correct principle of the single pump valve motion, as it supplied a mechanism that would give to the valve an additional impetus in the direction in which it had been moving, so as to uncover the ports at the opposite end after the piston stopped. The one bad feature was that it depended upon the momentum of the piston to turn the wristplate just a little over the center to release the spring.

This it would do all right at a fairly high speed, but at a slow speed the piston frequently failed to run past center and the pump stopped.

To remedy this it was thought that by inserting a spring the piston, in nearing the end of the stroke, would compress the spring without moving the valve and thus have a chance to complete its stroke. Then the spring would react and throw the valve. This idea was embodied in the second pump built, which is illustrated in Figs. 3 and 4, and which is now on exhibition in the Henry R. Worthington shops in Harrison, N. J.

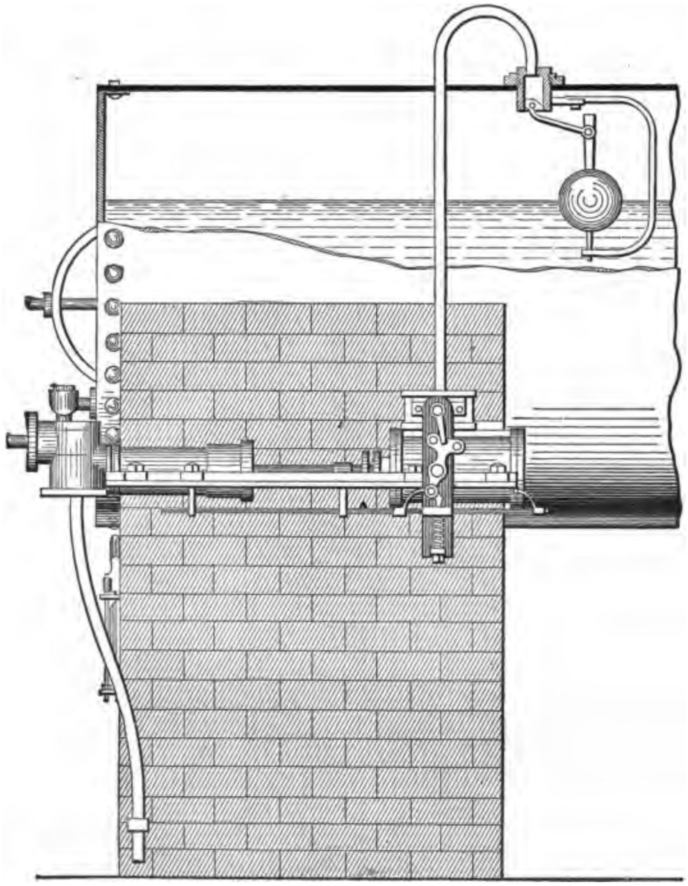


FIG. 2.—Automatic independent feed pump.

A tube *a*, with a spiral strip on the underside, was used in place of the valve rod and was acted upon by an arm *b* attached to the piston rod, which in moving back and forth gave an oscillatory motion to the tube. A rockshaft *d* located in the center of the tube formed the guide for a spiral spring which pressed against an arm *e* carrying a roller *f*. This roller, in oscillating around the shaft, engaged a triangular cam *g* at the end of arm *h*, which was secured to a rockshaft. The valve which was operated by this rockshaft, instead of working in line with the piston rod, passed over the ports in a transverse direction.

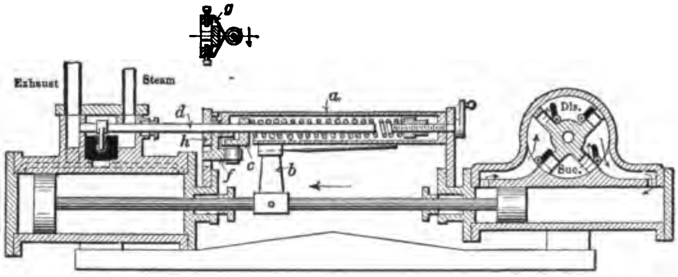


FIG. 3.—Single double-acting pump with spring-thrown valve, sectional view (Worthington, 1844).

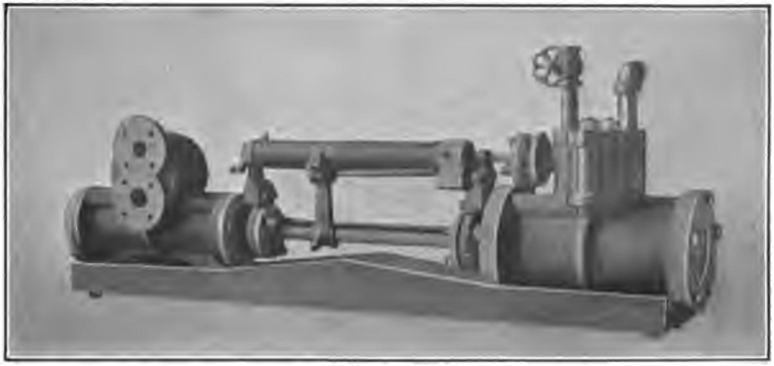


FIG. 4.—Outside view of pump shown in Fig. 3.

The spring action thus introduced, however, destroyed one of the good features of the first pump, in that it removed the positive action of the piston on the valve. In the first pump the piston closed the valve and thereby stopped its own motion; whereas, in the second pump the piston could complete its stroke and strike the head before the valve had time to reverse. This introduced an element of danger, should the valve motion fail to act promptly.

In this second pump, the valve remained in its extreme position until the roller had reached its own extreme position in which it had just ridden a little over the apex of the cam; then the valve was suddenly thrown across the ports from one extreme to the other. While many modern valve motions with steam-thrown valves employ this principle, still those that employ the principle of the first pump are considered superior, since the piston in closing its own valve forms a cushion and thereby limits its own stroke.

2. Steam-thrown Valve.—From a spring-thrown valve, it was only a step to a steam-thrown valve, shown in Fig. 5, which, however, was an awkward construction as the auxiliary piston *a* was stationary and the slide valve was loaded down with an auxiliary cylinder *b*, no means having been provided to allow the valve to find its own seat. Besides, the auxiliary piston had a piston rod projecting through a stuffing-box which was inaccessible. In describing this valve motion, Mr. Worthington said: "In my first arrangement, a rod protruded from the main steam valve as well as from the auxiliary, the latter not being shown. The object of this was either to compel the full motion of the main steam valve, or else the stoppage of the engine in case the auxiliary piston failed to work promptly. The auxiliary piston was acted upon by an auxiliary valve (not shown) that was placed on the main piston valve and was acted upon by a tappet arm through the medium of its own valve rod."

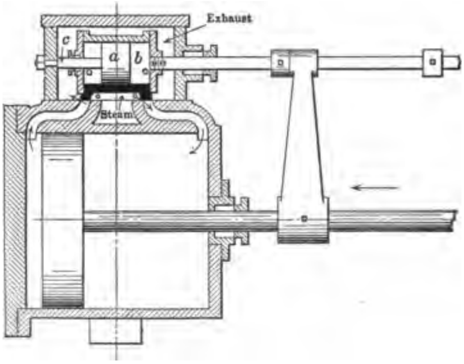


Fig. 5.—Steam-thrown valve (1847).

The illustration is not clear as it shows a "*D*" valve which necessitates a rockshaft and arm reversing the motion. An ordinary "*D*" valve always leads the piston, or, in other words, the piston follows its valve. Therefore, in order to complete the cycle, the piston, in moving, must move the valve in the opposite direction. However, a plain tappet arm is shown which moves the valve in the direction the piston is moving. Perhaps the difficulty was overcome by reversing the flow of steam, that is, admitting steam through the exhaust passage and exhausting through the steam opening, an arrangement that was adopted in the next type developed.

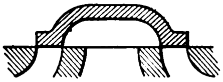


Fig. 6.—"*D*" valve.

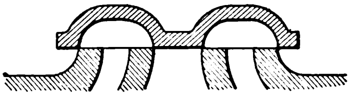


Fig. 7.—"*B*" valve.

3. "D" and "B" Valves.—A "*D*" valve, Fig. 6, has only one cavity in the center serving as an exhaust passage only, and the two bridges at the ends cover the ports in the center position. Its action is obvious. If it moves to the right, steam is admitted to the left-hand port, while the right-hand port is open to the exhaust. This causes the piston to move to the right; in other words, to follow the valve.

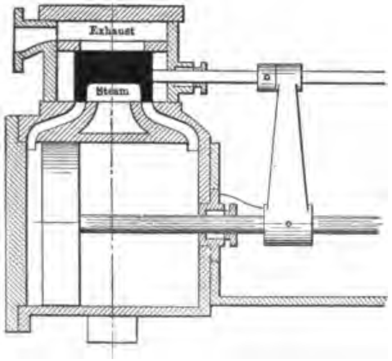


Fig. 8.—Balanced slide valve (1848).

A "*B*" valve, Fig. 7, has two cavities serving alternately as steam and as exhaust passages, with three bridges covering the ports. If the valve moves to the right, steam is admitted under the right-hand bridge to the right-hand port, while the left-hand port is opened to the exhaust, thus causing the piston to move in the opposite direction, viz., toward the left.

Difficulty resulted from the large size of piston required to handle an ordinary slide valve, and this suggested the balancing of the slide valve as shown in Fig. 8. In this balanced valve the direction of the steam was changed so as to have it enter at the exhaust port and escape into the steam chest which was thus converted into an exhaust chest. Plates were fitted over the blocks, at each end, forming a tight cavity wherein the valve was cushioned by the compression of the steam. A small valve was arranged to open at the proper time and destroy this dash-pot effect, so that the valve could be withdrawn more readily.

4. Exhaust-thrown Valve.—A modification of the auxiliary valve throw was made the subject of a patent in 1856 and is illustrated in Fig. 9. The steam valve was constructed in the form of a piston working in a small cylinder, which constituted the steam chest, the steam being admitted at the center and escaping at either end. As represented, the main piston has just completed the stroke from left to right, and the steam admitted through the central opening is about to pass to the right-hand end of the cylinder while the exhaust from the left-hand side is already commencing to escape through port *d*. This fills the space *c*, but the steam has no means of escape until it forces the piston valve toward the right far enough to uncover the opening *f*, which communicates with the exhaust pipe. By this time the steam port *b* is also open for steam. Thus the necessary motion of the steam valve is produced by the effort of the exhaust.

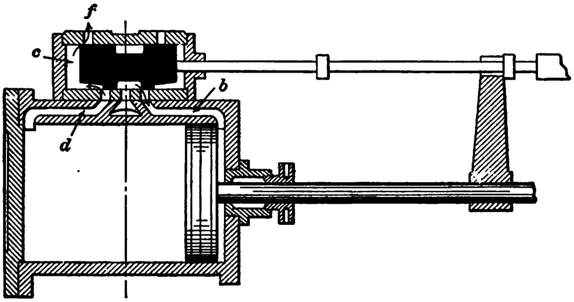


FIG. 9.—Exhaust-thrown valve (1856).

In the absence of any packing around the piston valve there must have been enormous short-circuit losses. No doubt these pumps worked poorly, especially when running slow, the steam leaking out rapidly at the joint between *c* and *f* before it had time to throw the valve.

5. Relief Valve Motion.—It may be surmised that, by watching such a pump at work, and seeing the piston stop, the idea must have occurred to the observer to devise some means by which the main piston could be made to jump a short distance at the critical moment and thus throw the valve full open in the opposite direction. This was accomplished by providing two water ports *a*, as shown in Fig. 10. The water piston, nearing the end of the stroke, uncovered the inner port, thus forming a communication between the two ends of the pump body, and relieving the load at the moment when the steam valve *b* had covered all the ports. The steam aided by momentum was then able to push the piston forward sufficiently to open the valve *b* fully.

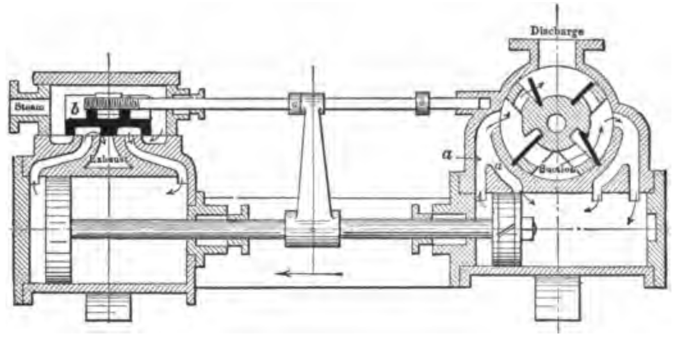


Fig. 10.—Piston pump with relief valve motion (1849).

This relief valve motion was further improved, as in Fig. 11, which shows the plunger and ring type such as is used at present.

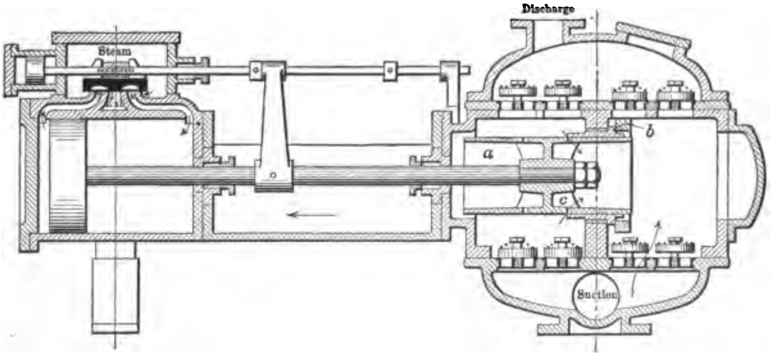


Fig. 11.—Plunger pump with relief valve motion (1850).

The plunger *a*, designed in the form of a long barrel working in a short ring *b* without any packing, was perforated near the diaphragm. The little holes *c* at the right-hand end, in traveling to the left of the plunger ring, would uncover and relieve the pressure. This produced the first really serviceable pump, as the relief of the pressure before reversal permitted the valves to seat quietly.

Coincidently should be mentioned another great improvement, namely, the employment of a number of small valves instead of one large water valve. These could not be expected to seat all at once, hence the shock imparted to the valve deck when one large heavy valve came down on its seat was eliminated. This principle of construction was first adopted in a pump used on board the steamer Washington in 1850. The valve was a plain india-rubber disk, half an inch thick, and working upon a central stem over a number of half-inch holes, the lift being not over a quarter of an inch. There were nine of these in each chamber of the pump.

Previous to this all pumps, whether attached or independent, had been provided with only one valve to each chamber, the earlier Worthington pumps being fitted with an oblong flat metal valve that rocked open and shut on a half hinge on one side. This was governed by a flat spring of hard brass, somewhat after the arrangement indicated in Fig. 10. A small obstruction under the heel of the valve would throw that end of the pump out of commission, so the demands for marine service, particularly for bilge pumping, called for improvement. The multiple valve scored a great success as the valve plates acted as a coarse strainer and all the rubbish drawn up from the bilge collected in the large suction chamber and could be removed from time to time as the pump showed signs of strangulation.

6. Plunger and Ring Type.—This type, already referred to in Fig. ~~10~~, 11 was an innovation and deserves a few comments. The construction makes it easy to renovate the working parts, or within reasonable limits to change the size of the plunger by merely withdrawing the plunger and packing ring, and substituting another. The packing ring is deep, without elasticity or provision for adjustment, and is carefully fitted to the plunger. The merits of such a design are evident when it is considered that there is little or no momentum to help out the stroke; hence, any inequalities or tight place in the packing must be avoided. As a cylinder or plunger always wears fastest at the middle of the stroke, if the packing were adjusted to fit the middle it would be likely to bind enough at the ends to hinder the motion.

Excellent as the relief valve motion was, as far as operation was concerned, there was more than one drawback to this type of pump. In the first place, the steam end was not complete in itself but depended upon the pump end where an essential part of the valve motion was located, namely, the relief ports. This prevented the steam end from being made as a unit. Secondly, the water end could be designed only as a plunger and ring pump, a solid piston pump, or one fitted with metallic piston rings, as soft packing evidently could not be used, for it would catch in the ports. So the problem did not appear to have been quite solved and Mr. Worthington strove for something different and better, finally developing one of the greatest inventions in pumping machinery—the duplex pump, which will be discussed later.

CHAPTER II

PERFORMANCE FACTORS

7. Efficiencies.—According to the character of the losses under consideration, a number of *efficiencies* are distinguished. The losses may be due to leaks past the valves, plungers or pistons and are then expressed by the *volumetric efficiency.* When incurred by passage of the liquid through the suction and discharge passages or valves they are expressed by the *hydraulic efficiency;* whereas, the mechanical losses occasioned by the friction of the moving parts are expressed by the *mechanical efficiency.* In the following these efficiencies and losses will be taken up in detail.

8. Volumetric Efficiency (E_v).—

$$E_v = \frac{G}{D}$$

where G = Actual volume discharged, generally stated in gallons per minute;

D = Plunger displacement, expressed in the same unit.

This efficiency gives information as to the loss of liquid due to defective piston packing, leaky stuffing-boxes or valves, the delayed closing of the valves and the amount of air carried into the pump body by the water. It is generally stated as loss in percentage of the displacement and is then called *slip.*

$$\text{Slip} = 1 - E_v$$

The slip is small in a new pump handling water that is reasonably free from air, actual tests having demonstrated that it rarely exceeds 2 per cent. in a plunger and ring pump. In packed pumps where there is no leakage past the plunger or piston it is less. Pumps handling large quantities of water have frequently shown slips of ½ per cent., no slip or even negative slip, the latter resulting from the tendency of the water to continue to flow after the plunger has stopped.

It is difficult to determine the slip accurately because there is no absolutely reliable method of measuring large flows of water. An error of 2 per cent. is to be expected in any case and this is more than the slip in a pump in good condition. A pump that shows excessive slip is in poor condition and should be overhauled. The cause of excessive leakage may be found in a worn plunger, an improperly packed piston, or defective valve service; that is, the valves may have worn or broken, or displaced by the breakage of a stem or the unfastening of a valve seat.

It may be said that slip is dependent upon the condition of the pump, the pressure and the speed. The first two, however, cannot be considered when determining the slip of pumps in general. Pumps for high pressures are designed differently from those for low pressures in order to keep the slip within reasonable limits. Therefore, it is only the speed that has a decided influence upon the slip. It is evident that the amount of leakage is constant per unit of time, and if expressed in percentage of the displacement,

$$\text{Slip} = \frac{\text{Leakage}}{\text{Displacement}}$$

From this it is evident that, with the rate of leakage constant, the slip will increase with decreasing displacement. If the slip at full capacity is 5 per cent., at half capacity it will be 10 per cent. and, if the speed be successively reduced, a point will be reached where the slip is 100 per cent., when the pump does not discharge any water at all. This law is utilized in testing small pumps. The discharge valve is closed and sufficient steam pressure is admitted to produce the normal water pressure. If the speed is then about 1 per cent. of normal, the pump is passed as satisfactory.

For convenience this speed is expressed by the number of single strokes per minute and is called *lost actions.* A 6-inch stroke duplex pump running at 65 feet per minute makes 260 single strokes per minute and will therefore pass with 3 lost actions. Here we have a pump in perfect condition showing a slip varying from 1 per cent. at full load to 100 per cent. at 1/100 load. In calculating the size of the plunger it is safe to assume a slip of 3 per cent., or a volumetric efficiency of 97 per cent. at normal speed.

9. Hydraulic Efficiency (E_h).—

$$E_h = \frac{h}{h_i}.$$

where h = Total head pumped against;
$h_i = h$ + hydraulic losses (indicated head).

The hydraulic losses consist of two parts:

A. Losses in passing from the well to the pump chamber;
B. Losses in passing from the pump chamber to the point where the discharge gage is attached.

The losses under A, due to the low velocity of the water and the short distance through which it travels, are usually small, and comprise:

a. Velocity head;
b. Entrance head;
c. Friction in the suction pipe;
d. Losses in the bends;
e. Loss in passing through the suction valves.

These losses will be taken up more in detail under "Suction Lift."

The losses under B comprise:

a. The loss in passing through the discharge valves, consisting of the head set up by the spring pressure and the friction through the valve seats. The valve is held to its seat by two forces, the water pressure in the force chamber, and the spring load, and it is clear that the valve cannot lift until the plunger has produced a pressure under the valve equal to the sum of the two. The proper spring tension should be such as to produce a pressure equal to from ½ to 1 per cent. of the discharge pressure. Friction through the valve seats will be very small, on account of the low velocity of the water, so that the total loss under this heading may be assumed to equal 1 per cent. of the water pressure.

b. The friction in the remaining passages, the velocity head, etc., all of which are negligible in a well-designed pump.

Friction of the water in the main discharge pipe is not included under hydraulic efficiency, because it is part of the useful work of the pump. If the point of consumption is at a great distance from the pump, the friction in the discharge pipe may be great as compared to the head, and if included in the hydraulic efficiency would make this very low. In fact, where the entire object of the pump is transportation in a horizontal direction and there is no static head, as is often the case in oil lines, the hydraulic efficiency would reduce to zero, which would be absurd. For this reason the discharge gage is always attached to the discharge pipe close to the pump and the *total head* is taken to equal the head, h_d (see Fig. 12), indicated by this gage, plus the vertical distance, h_s, from the center of this gage to the level of the water in the suction well.

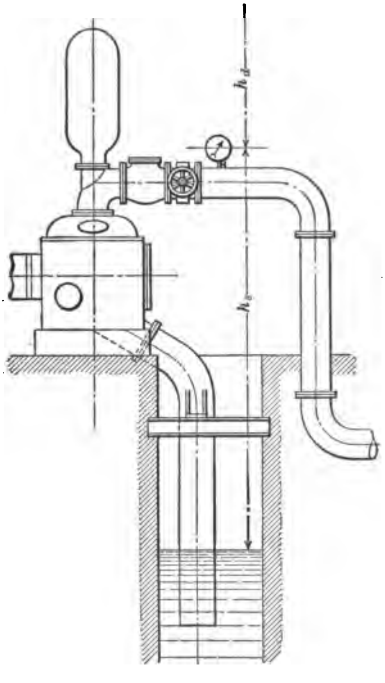


FIG. 12.—Measurement of total head.

It is difficult to measure the indicated head, h_i, from indicator diagrams of the pump end, owing to the disturbing influence of the inertia of the water augmented by long indicator pipes with elbows and valves. Best results are attained by screwing the indicator cock directly into the pump body at the center line and then attaching the indicator, which then will be in a horizontal position. This will provide a straight passage and produce fair diagrams. Fig. 13 shows actual water diagrams taken at high speed. This efficiency is generally included in the expression, for the mechanical efficiency.

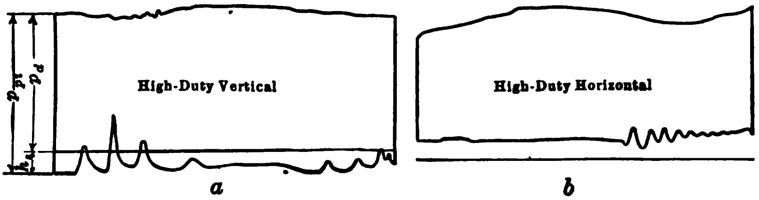


FIG. 13.—Actual water diagrams.

10. Indicated Pump Efficiency (E_{pi}).—This efficiency is the ratio of the power developed in raising the water to the power developed in the pump end as disclosed by the indicator diagrams.

$$E_{pi} = \frac{HP_w}{HP_i}$$

where HP_w = Water horsepower;

HP_i = Indicated pump horsepower.

The value of HP_w is calculated from the amount of water, G, actually pumped and the total head h; whereas, HP_i is the horsepower calculated from the displacement D and the indicated head h_i. The indicated pump efficiency can then be expressed by

$$E_{pi} = \frac{Gh}{Dh_i}$$

It is also the product of the hydraulic efficiency and the volumetric efficiency:

$$E_{pi} = E_h \times E_v$$

and comprises all losses in the pump end.

11. Mechanical Efficiency (E_m).—This is the ratio of the power developed in the pump end to that developed in the steam end, as disclosed by the indicator diagrams. It is expressed by

$$E_m = \frac{HP_{pi}}{HP_{si}}$$

If the hydraulic efficiency is included as explained under 9, then the mechanical efficiency is the ratio of the power developed in raising the water, to the power developed in the steam end, viz.:

$$E_m = \frac{HP_w}{HP_{si}}$$

The mechanical efficiency gives information as to the mechanical friction in the mechanism transmitting the power to the pump and can be determined only by actual experiments with the different types. It depends upon the size of the pump and the service. In Table 1 mechanical efficiencies are given for various types of direct-acting pumps, the service being roughly considered. The pumps are also classified according to the length of stroke which is, to some extent, a measure of the size.

TABLE 1.—SPEEDS AND MECHANICAL EFFICIENCIES

Stroke, in.	Normal speeds						E_m (including hydraulic efficiency)				
	Piston, plunger, outside packed				Pressure and wet vacuum pumps	Boiler feed	Mechanical efficiencies, per cent.				
	Simple cyl. and compound		Triple expansion			Thick liquids	Piston, plunger and ring, 150 lb.	Outside packed, 300 lb.	Press. pumps		
	R.p.m.	Ft. per min.	R.p.m.	Ft. per min.	R.p.m.	Ft. per min.			1000 lb. per sq. in.	3000 lb. per sq. in.	
3	80	40	......	.....	32	20	50	47	45	39	Also for thick liquids
4	75	50	......	.....	40	25	55	52	50	43	
5	72	60	......	.....	48	30	60	57	54	47	
6	65	65	......	.....	52	33	65	61	58	51	
8	56.2	75	......	.....	60	38	70	66	63	55	
10	48	80	54	90	64	40	75	71	67	58	
12	45	90	50	100	72	45	77.5	74	70	60	
15	40	100	44	110	80	50	80	76	72	62	
18	36.7	110	40	120	88	55	82.5	78	74	64	
24	30	120	32.5	130	96	60	85	81	77	66	
30	25	125	28	140	100	63	85	81	77	66	
36	22.5	135	25	150	108	68	87.5	83	79	68	
48	18	145	20	160	116	73	90	85	81	70	
60	15	150	17	170	120	75	90	85	81	70	

The first column under E_m gives the mechanical efficiencies for piston pumps and plunger and ring pumps for pressures up to 150 pounds. The second column gives the values for outside centerpacked plunger pumps for pressures up to 300 pounds. The next two columns give two sets of values, one for pressure pumps at 1000 pounds and the other for pressure pumps at 3000 pounds. For intermediate pressures the mechanical efficiency may be found by interpolation. The values in the last col-

normal [illegible]
for [illegible] run [illegible]
[illegible]

umn can also be used for thick liquids such as crude oil. molasses, tar, etc.

12. Total Efficiency (*E*).—This efficiency comprises all losses in the driving mechanism and the pump and indicates the economy of the whole outfit. It is expressed by

$$E = E_h \times E_v \times E_m$$

13. Thermal Efficiency (E_t).—This is the ratio of the heat utilized to the heat supplied and can be expressed by the following formula:

$$E_t = \frac{42.41 \times HP \times 60}{S(H - h)}$$

where HP = Horsepower;

42.41 = Heat equivalent of one horsepower, in B.t.u. per minute;

S = Steam consumed in pounds per hour;
H = Total heat in 1 pound of steam at initial pressure;
h = Total heat in 1 pound of feed-water.

The point at which the temperature of the feed-water is to be measured depends upon whether it is heated by the exhaust of the main engine, by the exhaust of some auxiliary pumps or by other means. All these points should be considered beforehand and the contract should be drawn up in such a manner as to be perfectly clear in this respect. This will relieve the testing engineer of all responsibility and may save him much disagreeable discussion.

14. Suction Lift.—The suction action of a pump is produced by the atmosphere resting on the surface of the water in the well and forcing the water into the pump chamber after the plunger has created a vacuum in the latter. The height of water column sustained by the atmosphere depends upon the temperature of the water and the various values are given in Table 2. To lift water to these heights by suction would require a perfect vacuum which is not possible to attain because the water boils before this condition is reached and the space becomes filled with vapors. The boiling point varies with the temperature and the absolute pressure corresponding to the boiling point gives a value for the theoretically attainable suction lift.

TABLE 2.—SUCTION LIFTS

Temp. deg. F.	One atmosphere in feet of water column	Abs. press. in feet of water column at which water boil.	Theoretical suction lift	Actual suction	
				Lift	Head
60	34	0.6	33.4	22	....
100	34.2	2.2	32	13	....
140	34.4	6.8	27.6	4	....
160	34.5	11.2	23.3	0	0
180	34.7	18	16.7	....	5
190	35	22.3	12.7	....	7
212	35.4	35.4	0	....	12

15. Velocity Head (h_v).—A certain amount of energy must be expended to set the water in the suction pipe in motion, this energy being equal to the weight of the water multiplied by a certain head in feet. This head is called "velocity head" and is found by the well-known formula of gravitation

$$h_v = \frac{v^2}{2g}$$

where v = Velocity in feet per second;
$2g$ = Acceleration due to gravity = 64.4. 32.2

If we assume a velocity of 3 feet per second this becomes

$$h_v = \frac{9}{64.4} = 0.14 \text{ ft. (water column)}$$

16. Entrance Head (h_e).—At the point of entrance there is an additional loss of head called "entrance head" which depends upon the shape of the suction pipe end and may assume the following values:

End of suction pipe—square, $h_e = \frac{h_v}{2}$.

End of suction pipe—funnel shaped, $h_e = 0$.

End of suction pipe provided with a foot valve, or strainer, $h_e = h_v$.

Assuming the first case, with the same velocity as before, $h_e = 0.07$ feet.

17. Friction Head (h_f).—The friction in the suction pipe is represented by h_f. To arrive at some average value assume a pump having a 12-inch suction, 30 feet long and a capacity of 1000 gallons per minute. The friction will then be 4 feet in 1000 feet or $h_f = 0.012$ feet in 30 feet; see Tables 3, 4. and 5 (section 33).

18. Losses in Bends (h_b).—Generally these losses will be small because the suction pipe is usually short and direct, with perhaps one or two bends. If there are many bends the losses through them can be determined according to the following rules:

Long radius, $h_b = \frac{h_v}{4}$

Radius = Diameter of pipe, $h_b = \frac{h_v}{2}$

Sharp bend, $h_b = h_v$

In the latter case it is assumed that the whole velocity is destroyed and must be set up again in some other direction. Assuming three bends, each of a radius equal to the diameter of the pipe,

$$h_b = 3h_v \div 2 = 0.21 \text{ ft. (water column)}$$

19. Loss through Suction Valves (h_s).—This is made up of the pressure necessary to open the valve against the spring pressure and the friction through the valve seat. It may be assumed as ½ pound per square inch or,

$$h_s = 1.15 \text{ ft. (water column)}$$

Summing up, the total *hydraulic losses* from the suction well to the pump chamber will, under the assumed conditions, be:

$$\begin{aligned} h_v &= 0.14 \\ h_e &= 0.07 \\ h_f &= 0.012 \\ h_b &= 0.21 \\ h_s &= 1.15 \\ &\cdots\cdots\cdots\cdots \\ &1.582 \text{ ft. (water column)} \end{aligned}$$

These must be deducted from the theoretical suction lift, but do not explain the difference between the theoretical and actual suction lifts as given in Table 2. The reason for this will next be explained.

In dealing with this question many authors lay stress on the inertia of the water in the suction pipe which, as they claim, comes to rest at the end of every stroke and must be set in motion again at the beginning of the succeeding stroke. Complicated formulas are then developed and curves drawn to show the effect of this inertia at every point of the stroke. While the theories thus developed may be correct and may have some value in a single, single-acting pump without an air chamber, a relatively unimportant type, they do not apply to pumps with a vacuum chamber, and especially not to multiplex pumps, where the water in the suction pipe is in uniform motion.

20. Presence of Air.—Water under atmospheric conditions contains about 2 per cent. air in solution, and, if subjected to a vacuum, this air will expand. Under a suction lift of 17 feet it would expand to 4 per cent., and with a suction lift of 28 feet, which is equivalent to an absolute pressure of 6 feet of water, it would expand to

$$\frac{34}{6} \times 2 = 11.3 \text{ per cent., and so on.}$$

The air would then fill 11.3 per cent. of the water cylinder and cut down the capacity accordingly, which in the case of a direct-acting pump would cause serious disturbances. In practice, however, it is not as bad as this, because the water passes through the short suction pipe so quickly that the air has not time to become liberated from the water. This explains why long suction pipes are a detriment. The effect of this air cannot be determined theoretically, because the amount liberated in a given time and under such complicated conditions cannot be calculated accurately.

Fig. 14 shows the actual suction lifts attainable with water at various temperatures, *a* being the actual suction lift under ordinary conditions, *m* the maximum suction lift under favorable conditions, such as may prevail during a test, and *t* the theoretical suction lift. All the curves are based on the zero line which represents the atmospheric pressure. The points to the right of this line indicate "suction lift," and those to the left "suction head." The line marked "absolute zero" gives the equivalent of the atmospheric pressure in feet of water at sea level, which, of course, increases with the temperature.

This diagram is based on a velocity of from 3 to 4 feet per second and a short suction pipe with one bend. If the suction pipe is long, the velocity high, or there are many bends or other obstructions, such as a foot valve, strainer or surface condenser, the losses must be estimated and deducted from the available lift indicated by curve *a*.

The lines *a* and *m* are not extended beyond the 212 degree line, because water hotter than this cannot exist except under pressure and the effect of this pressure is added to the static head. Therefore, under the conditions stated, a head of 12 feet is sufficient for any temperature.

In **condenser work** where hot water is handled, that is perfectly free from air, a head of a few feet only is required to make the water flow from under a high vacuum to the suction chamber. Thus the equivalent of a very high suction lift, 28 or 30 feet, is frequently realized. **Thick liquids** should always flow to the pump by gravity or, in other words, under a head.

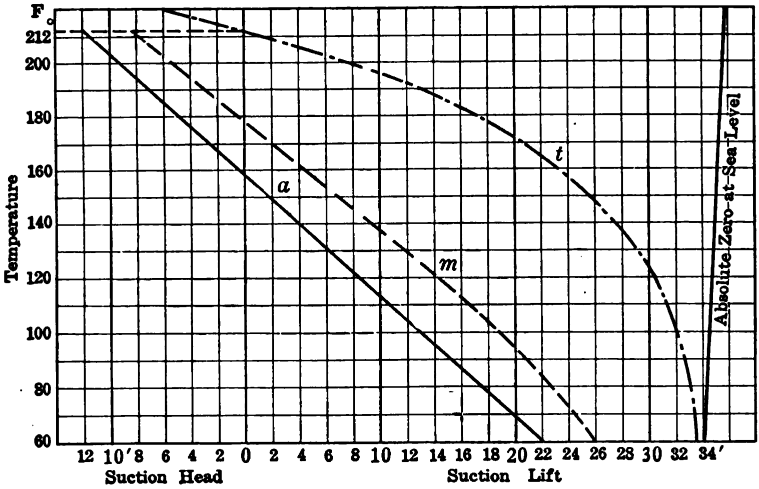


Fig. 14.—Suction lift diagram.

21. Air and Vacuum Chambers.—An air chamber is a vessel of suitable material, generally cast iron, having a single opening at the bottom and attached to the discharge pipe. When attached to the suction pipe it is called a vacuum chamber or suction air chamber. Its object is to provide an elastic element in the pipe line, taking up the shocks and irregularities and inducing a uniform flow in the pipe. Air chambers are a necessity in single, single-acting pumps, in duplex single-acting pumps, and in crank-and-flywheel pumps of any type. They may be dispensed with in duplex double-acting low-service pumps (75 pounds pressure), in small duplex pumps for general service (up to 10 × 6 × 10 inches), and in small slow-running pumps, such as the deep-well bucket type.

The volume of the air chamber should be six to eight times the displacement in single direct-acting or crank and flywheel pumps, and three to four times in duplex pumps.

Air chambers should always be provided with glass gages and a petcock on top as a means for letting out any excess of air and keeping the water level at the proper height. In a closed system, such as with elevators, the water does not carry much air, in which case some air-charging device or an air compressor must frequently be provided. The air-charging device, Fig. 15*a*, utilizes the varying pressure in one of the pump chambers to compress air into the air chamber. A vertical tube, *t*, is attached to the side of the pump body and is in communication with the inside of one of the pump chambers. The upper end carries two check valves, one for suction and one for discharge, communicating with the air chamber. The pulsations in the pump chamber produce similar pulsations in the tube, thus forming an

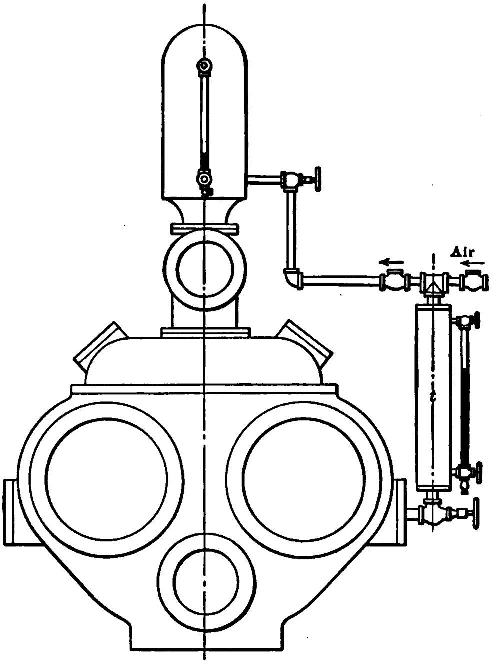


Fig. 15*a*.—Air charging device.

effective air compressor, without danger of the air entering the pump chamber.

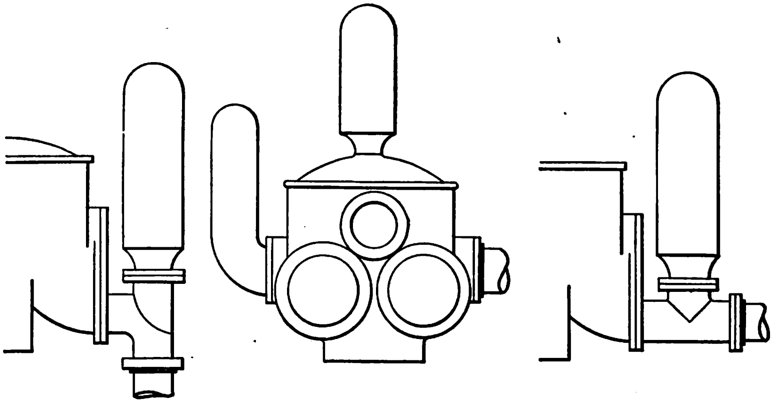


FIG. 15*b*.—Good. FIG. 15*c*.—Good. FIG. 15*d*.—Bad.

FIGS. 15*b*, 15*c* and 15*d*.—Locations of vacuum chamber.

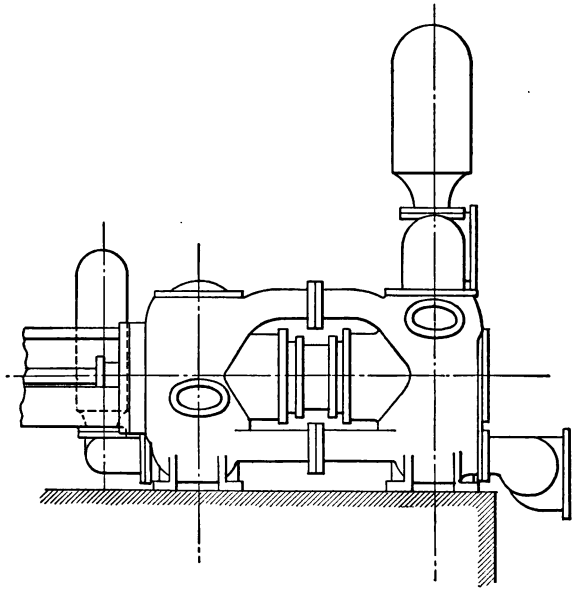


FIG. 16.—Vacuum chamber on run-around pipe.

The location of the vacuum chamber should receive careful attention. It should usually be placed on the dead end of a tee opposite to where the suction pipe is attached (Fig. 15*b*), or as in Fig. 15*c*. If placed as in Fig. 15*d*, much of the cushioning effect is lost. Another good place, especially on large pumps, is on top of the "run-around pipe," which connects the inboard ends of the two suction chambers of a duplex pump and permits uninterrupted flow from one suction chamber to the other. Such an arrangement is shown in Fig. 16.

The air liberated from the water during the suction period finds its way into the vacuum chamber until it has displaced all the water and the chamber is full of air. From then on, it enters the pump chamber as if there were no vacuum chamber. In artesian-well service, where the water is impregnated with air, it is often necessary to remove this from the vacuum chamber by means of a vacuum pump.

22. Capacity.—The capacity of a single, double-acting pump can be calculated theoretically by the following formula:

$$D = \frac{a \times P \times 12}{231}$$

or,

$$D = 0.0408\, d^2P$$

where D = Displacement of one double-acting plunger, in U. S. gallons per minute;
a = Area of plunger in square inches;
d = Diameter of plunger in inches;
P = Piston speed in feet per minute.

From this the displacement of any other type is readily obtained. For a

Single, single-acting pump divide D by 2,
Duplex, double-acting pump multiply D by 2,
Triplex, single-acting pump multiply D by $3/2$,
Triplex, double-acting pump multiply D by 3, and so on.

From the displacement of the plunger or piston the displacement of the piston rod must be deducted. In most cases this extends through the inboard end of the pump and is attached to one end of the plunger. It then forms a single-acting plunger and its displacement can be calculated by the formula given. If a tail rod is attached to the other end the displacement of this must also be deducted. End-packed pressure pumps and outside center-packed pumps with a crosshead between the two pump bodies (pressure pumps with side rods) have no piston rod and no allowance is to be made in this case.

To find the actual capacity or delivery of a pump, the slip must be deducted from the displacement. Under the head of "Volumetric Efficiency" this was shown to be very small, but in calculating sizes it is best to allow ample margin, and a slip of 3 per cent. should be assumed.

It is well to note in passing that there are two kinds of gallons in use, namely: the *U. S. gallon* of 231 cubic inches, and the *Imperial gallon*, used in Great Britain and her colonies, which is equivalent to 277 cubic inches, or 20 per cent. larger than the *U. S. gallon.*

If the capacity is given and the diameter is sought, the formula for one double-acting plunger may be transposed into

$$d^2P = 24.51D$$

Calculate d^2P and then by inserting some average piston speed find the corresponding diameter. After the diameter is known approximately, select some suitable stroke which fixes the proper speed (see Table 1) from which the exact diameter can then be calculated.

Example (1).—Find the plunger diameter for a duplex pump of 24-inch stroke, running at 120 feet per minute. The capacity G is 1300 gallons per minute, and the diameter of the piston rod (d_r) is 2¾ inches.

Solution.—The displacement (D) of one side allowing 3 per cent. for slip, is

$$D = \frac{G}{2} + 3 \text{ per cent.} = 670 \text{ gal. per min.}$$

To this must be added the displacement of the piston rod (D_r),

$$D^r = \frac{0.0408}{2} d_r^2 P$$

$$= \frac{0.0408}{2} \times 2.75^2 \times 120 = 18.5 \text{ gal. per min.}$$

The displacement of the plunger will then be

$$D_p = D + D_r = 688.5 \text{ gal. per min.}$$

As previously stated $d^2P = 24.51D$
whence

$$d^2 \times P = 24.51 \times 688.5 = 16{,}875$$

and,

$$d^2 = \frac{16{,}875}{120} = 140.6$$

Therefore, the diameter of the plunger

$$d = \sqrt{140.6} = 11.85$$

or say 11⅞ inches.

Example (2).—A vertical differential-plunger pump having two single-acting plungers 45 inches in diameter is running at a speed of 150 feet per minute. What is the capacity? Allow 3 per cent. for slip. There is no piston rod.

Solution.—

$$D = 2 \times \frac{0.0408}{2} d^2P$$

$$D = 2 \times 0.0204 \times 45^2 \times 150 = 12{,}394 \text{ gal. per min.}$$

From this 3 per cent. must be deducted for slip; hence to find the capacity,

$$G = 12{,}394 \times 0.97 = 12{,}022 \text{ gal. per min.}$$

The capacity of large pumping engines is generally stated in gallons per 24 hours, and since there are 1440 minutes in 24 hours, in this particular case the capacity is

$$12{,}022 \times 1440 = 17{,}311{,}700 \text{ gal. per 24 hr.}$$

23. Speeds.—In determining the normal speeds of reciprocating pumps, two points must be considered: the average piston speed and the number of reversals.

Obviously, a small pump, that is, one with a short stroke, can run with more frequent reversals than a large pump having a long stroke, but a long-stroke pump will run with a higher piston speed than a short-stroke pump.

To designate the number of reversals, it has been proposed to state the number of revolutions per minute, rather than the number of strokes. There being two plungers in a duplex pump, it was always more or less doubtful what was meant by the number of strokes. On the other hand, the number of revolutions designates one complete cycle of the reciprocating parts, and is readily obtained by counting the movements of any moving part, such as one of the crossheads.

Table 1 gives the proper speeds of direct-acting pumps, from which it will be noted that the speed increases with the stroke, while the revolutions per minute decrease. Slightly higher speeds are given to triple-expansion, than to compound pumps.

Pressure pumps, built for 300 pounds and over, should run at about 80 per cent. of the speeds for regular pumps. This is necessary because of the greater danger due to shocks and water hammer.

Boiler-feed pumps should run at half speed, so that in emergencies they can be speeded up. Furthermore, a slow-running pump is not subjected to any shocks and is not liable to be deranged so easily—an essential feature in such important service.

Pumps handling thick liquids such as syrup, molasses, heavy oil, etc., which cannot be made to flow fast, should also run slowly.

24. Power.—The horsepower of a pump can be calculated by the well-known formula:

$$HP = \frac{plan}{33000}$$

where p = Mean effective pressure in pounds per square inch;
l = Length of stroke in feet;
a = Area of piston in square inches;
n = Number of effective strokes per minute.

The "effective" strokes have been used to guard against errors in dealing with single-acting plungers, which make only one effective stroke per revolution. Relative values for n are as follows:

Single, single-acting.......... $n =$ r.p.m.
Single, double-acting......... $n = 2 \times$ r.p.m.
Duplex, double-acting........ $n = 4 \times$ r.p.m.

In applying the above formula to single, double-acting pumps the piston speed can be substituted for $l \times n$.

Depending upon whether p in the formula means the indicated mean effective pressure as determined by the indicator or the water pressure as determined by the gages, also whether the volumetric efficiency is taken into account, five different values must be distinguished. These will be taken up in order.

25. Indicated Horsepower of the Steam End (HP_{si}).—

$$HP_{si} = \frac{p_{si} l a_s n}{33000}$$

where p_{si} = Indicated mean effective pressure in the steam cylinder in pounds per square inch;
a_s = Area of the steam piston in square inches.

This horsepower is determined by the indicator or is calculated from the theoretical diagram.

26. Brake Horsepower (HP_b).—This is the power actually transmitted to the pump. In direct-acting pumps it cannot be measured and has no meaning.

$$HP_b = HP_{si} \times E_{ms}$$

where E_{ms} is the mechanical efficiency of the steam end.

27. Indicated Horsepower of Pump End (HP_{pi}).—

$$HP_{pi} = \frac{p_{pi}\, l\, a_p\, n}{33000}$$

where p_{pi} = Indicated mean effective pressure in the pump cylinder in pounds per square inch;

a_p = Area of the plunger in square inches,

$$HP_{pi} = HP_b \times E_{mp}$$

where E_{mp} = The mechanical efficiency of the pump end.

As previously explained, the indicated pump mean effective pressure is difficult to obtain; therefore it is preferred to use the pump horsepower.

28. Pump Horsepower (HP_p). —

$$HP_p = \frac{p\, l\, a_p\, n}{33000} = \frac{Dwh}{33000}$$

where p = Total water pressure in pounds per square inch;

h = Total head in feet = 2.31 p, at 62 deg. F.;

D = Displacement of plunger in gallons per minute;

w = Weight of 1 gallon in pounds at the temperature of the liquid = 8.33 pounds at 62 deg. F.

The values, p and h, are measured by the discharge gage, to which reading is added the height of the discharge gage above the water level in the suction well (see Fig. 12).

$$HP_p = HP_{si} \times E_m$$

where E_m = Mechanical efficiency, including the hydraulic losses and the volumetric efficiency.

29. Water or Developed Horsepower (HP_w).—This expresses the useful work done by the pump.

$$HP_w = HP_p \times E_v = \frac{Gwh}{33000}$$

where G = Amount of water actually discharged, in gallons per minute;

E_v = Volumetric efficiency.

Example.—What are the horsepowers of the pump in Example 2, section 22, where G = 18,032 gallons per minute? Assume the total head, h, to be 130 feet.

Solution.—

$$HP_w = \frac{Gwh}{33000} = \frac{18032 \times 8.33 \times 130}{33000} = 592$$

$$HP_p = \frac{592}{0.97} = 610$$

To find the indicated horsepower of the steam end, assume a stroke of 60 inches and from Table 1 a corresponding mechanical efficiency of 85 per cent. (this includes the hydraulic losses). Then,

$$HP_{si} = \frac{610}{0.85} = 720$$

30. Pipe Sizes.—The pump is always furnished complete by the manufacturer, so that the customer needs only to connect the suction opening to the suction pipe and the discharge opening to the discharge pipe. In a duplex pump the suction passages of the two sides connect into a single passage terminating in one opening, and the discharge likewise has one opening. The velocity in the suction pipe should not exceed 240 feet per minute, and that in the discharge pipe 300 feet. The pipe diameters can be expressed by the following:

$$\text{Suction diameter} = \sqrt{0.1\,G}$$
$$\text{Discharge diameter} = \sqrt{0.08\,G}$$

where G = Gallons per minute.

31. Starting Pipe.—Large pumps are provided with an extra opening on the main discharge, which is fitted with a valve and piped back to the suction well. This is called the *starting pipe*, and assists greatly in starting by allowing the air to escape and relieving the force chamber of all pressure until the capacity has increased to such an extent as to open the check valve, when the starting valve is closed. The diameter of this starting valve is made equal to one-eighth of the diameter of the discharge pipe plus ½ inch.

32.—Priming Pipes.—To facilitate starting some means should be provided for filling the pump with water, for if the pump and the suction pipe are empty the former will work as an air pump and will remove the air only in a very imperfect manner. If, however, the pump chambers are filled with water the clearance spaces are made harmless, and the pump will easily pick up.

If the end of the suction pipe is fitted with a foot valve it is necessary only to connect the suction pipe to some water supply, such as the city main, and by admitting this water the suction pipe and the pump chambers will fill and drive the air out through the starting pipe. If there is no foot valve, each chamber must be connected separately to some such water supply. Often it is sufficient to connect each chamber to the force chamber, because the water contained in the force chamber and the discharge pipe will fill the pump chambers. Even though they are not completely filled by this means, if the priming pipes are left open and the pump is started, the air will work out through the priming pipes much better than if it had to force its way through the discharge valves.

An excellent contrivance for removing the air is to fit each pump chamber with an air vent valve, as shown in Fig. 17. The valve is closed by a handwheel and, if the stop is withdrawn, the valve is free to open for venting the pump chamber but will close on the suction stroke.

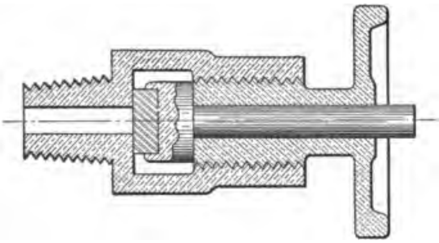


FIG. 17.—Air-vent valve.

To protect the discharge pipe a relief valve is frequently attached. This should have a diameter equal to one-half that of the discharge pipe.

33. Friction in Pipe Lines.—As this book is intended to be a treatise on direct-acting pumps only, a lengthy discussion of pipe-line friction would be out of place. The subject, however, is of such importance that brief mention should be made of the excellent book on "Pipe Friction Hydraulic Tables," by Hazen and Williams, which is published by John Wiley & Sons. These have been condensed into three small tables which will be found handy if the original work is not available. They show the flow of water in pipes 1000 feet long, as computed by the Hazen-Williams formula which is:

$$V = 1.32\,C \times r^{0.63}S^{0.54}$$

Where V = Velocity of water in feet per second;
C = An empirical coefficient;
r = Hydraulic radius in feet $= \frac{d}{4}$ for pipes running full;
S = Hydraulic slope $\frac{h}{l} = \frac{head}{1000}$.

TABLE 3.—FRICTION HEAD IN FEET PER 1000-FOOT LENGTH OF PIPE

NOTE.—Figures in this table give the friction head in feet per 1000-foot length of pipe based on factor $C = 100$ for pipes of condition or age represented opposite factor $C = 100$ in Table 3.

$C = 100$	Size of pipe in inches												
	Standard wrought-iron pipe								2	2-½	3	4	5
	⅛	¼	⅜	½	¾	1	1-¼	1-½					
Inside diameter d =	0.27	0.364	0.494	0.623	0.824	1.048	1.38	1.611					
At constant head capacity is proportional to $d^{2.63}$ =	0.032	0.07	0.157	0.289	0.601	1.13	2.33	3.5	6.19	11.1	18	38.3	69
At constant capacity head is proportional to $\frac{1000}{d^{4.87}}$ =	588,000	137,000	31,000	10,000	2,560	796	208	98	34.2	11.5	4.75	1.17	0.395
U. S. gallons per minute = G													
0.2	62						For other values of C multiply figures in table by A						
0.5	345	78											
1	1210	280	64	21									
1.5		600	138	45									
2		1030	233	74	19								
3			490	158	41	12.6							
4			840	270	70	21.4	5.7	2.62					
5			1260	410	105	32.5	8.4	3.98					
6				570	147	45.5	12	5.6	2				
8				980	250	78	20.3	9.5	3.3	1.1			
10				1470	380	117	30.5	14.3	5	1.7	0.7		
15					800	250	65	30	10.7	3.6	1.5		
20					1360	420	111	52	18.2	6.1	2.5	0.62	
30						890	235	110	38.4	12.9	5.4	1.32	0.44
40						1520	400	188	66	22	9.1	2.23	0.75
50							600	284	99	33.2	13.8	3.39	1.13
60							850	396	139	46.5	19.2	4.72	1.59
80							1450	680	237	79	32.8	8.1	2.71
100								1020	358	120	49.6	12.2	4.11
150									760	256	105	26	8.7
200									1290	431	178	44	14.8
300										920	380	93	31.4
400										1560	650	160	54
500											980	240	81
600												337	113

At constant diameter, head is proportional to $G^{1.85}$ and capacity to $h^{.54}$.

$y : 337 = 4.75 : 1.17$

$y = \frac{337 \times 4.75}{1.17} = 1370$

TABLE 4.—FRICTION HEAD IN FEET PER 1000-FOOT LENGTH OF PIPE

NOTE.—Figures in this table give the friction head in feet per 1000-foot length of pipe based on factor $C = 100$ for pipes of age represented opposite factor $C = 100$ in Table 3.

$C = 100$		Size of pipe in inches												
		6	8	10	12	16	20	24	30	36	42	48	54	60
At constant head capacity is proportional to $d^{2.63}$ =		111	237	427	690	1,469	2,645	4,266	7,683	12,420	18,630	26,430	36,060	47,650
At constant capacity head is proportional to $\frac{100 \text{ million}}{d^{4.87}}$ =		16,260	4,000	1,350	556	137	46.1	19	6.4	2.63	1.12	0.652	0.367	0.219
U. S. gallons per minute = G	35	0.24						For other values of C multiply figures in table by A						
	42	0.33												
	56	0.57	0.13											
	70	0.86	0.22	0.07	0.03									
	105	1.84	0.43	0.16	0.07									
	140	3.1	0.77	0.26	0.11	0.026								
	210	6.6	1.62	0.55	0.22	0.056								
	280	11.3	2.76	0.93	0.38	0.095	0.032							
	350	16.9	4.18	1.41	0.58	0.143	0.049	0.02						
	420	23.8	5.9	1.97	0.81	0.201	0.068	0.03						
	560	40.4	9.9	3.38	1.38	0.34	0.115	0.05						
	700	61	15.1	5.1	2.1	0.52	0.174	0.072	0.024					
	1,400	222	54	18.4	7.6	1.87	0.63	0.259	0.087	0.036				
	2,100		116	38.6	16.0	3.98	1.33	0.55	0.184	0.076	0.036			
	2,800			66	27	6.8	2.28	0.93	0.315	0.129	0.061	0.032	0.018	0.011
	3,500				41	10.2	3.43	1.41	0.476	0.196	0.092	0.048	0.028	0.016
	4,200				58	14.3	4.81	1.97	0.67	0.274	0.129	0.068	0.038	0.023
	5,600				99	24.2	8.2	3.38	1.13	0.467	0.22	0.115	0.065	0.039
	7,000				150	36.8	12.4	5.1	1.72	0.71	0.332	0.174	0.098	0.059
	10,500					78	26.2	10.8	3.64	1.49	0.7	0.37	0.21	0.125
	14,000						44.8	18.3	6.2	2.55	1.21	0.63	0.354	0.212
	21,000							39	13.2	5.4	2.56	1.33	0.75	0.449
	28,000								22.4	9.2	4.35	2.28	1.28	0.76
	35,000									13.9	6.6	3.44	1.94	1.16
	42,000									19.6	9.2	4.8	2.71	1.62
	56,000										15.7	8.2	4.61	2.78
	70,000											12.4	7	4.19

At constant diameter, head is proportional to $G^{1.85}$, and capacity to $h^{.54}$.

TABLE 5.—GIVING COEFFICIENT OF CONDITION C, FACTOR A, AND CORRESPONDING YEARS OF SERVICE IN SOFT, CLEAR, UNFILTERED RIVER WATER

C	$0.54\sqrt{\frac{100}{C}} = A$	Diameter of pipe in inches															
		⅛-1½	2-3	4	5	6	8	10	12	16	20	24	30	36	42 48	54	60
		Condition of pipe		Years of service of cast-iron pipe													
140	0.54	Very smooth and straight W. I.	Very smooth and straight W. I., brass, tin, etc.	oo	oo	oo	oo	oo	oo	oo	oo	oo	oo	oo	oo	oo	oo
130	0.62	....	Ordinary straight brass, tin, etc.	o	o	o	o	o	o	o	o	o	o	o	o	o	o
120	0.71	Smooth new wrought iron....................		4	4	4	5	5	5	5	5	5	6	6	6	6	6
110	0.84	....		....	....	....	10	10	10	11	11	11	12	12	12	12	12
100	**1**	**Ordinary wrought iron**....................		**13**	**14**	**15**	**16**	**17**	**17**	**18**	**19**	**19**	**19**	**20**	**20**	**20**	**20**
90	1.22	....		....	....	....	....	....	26	27	28	29	30	30	30	31	31
80	1.52	Old wrought iron.......	....	26	28	30	33	35	37	39	41	42	43	44	45	46	47
60	2.58	....	Very rough............	45	50	55	62	68	....	....	....	....	....	....	....	....	....
40	5.46	Very rough...........	Badly tuberculated......	75	87	95	....	....	....	....	....	....	....	....	....	....	....

"oo" means very best and new cast-iron straight pipe; "o" means good new cast-iron pipe.

The friction heads are given for $C = 100$ only; for other values of C multiply by $A = \sqrt[0.54]{\frac{100}{C}}$.

APPROXIMATE RULES FOR INTERPOLATING

Rule 1.—*At constant head, capacity is proportional to* $d^{2.63}$ (roughly $d^{2.5}$)

Example.—A 4-inch pipe discharges 60 gallons. How much would a 2-inch pipe discharge under the same conditions?

$$\text{Capacity} = \frac{60 \times 6.19}{38.3} = 9.7 \text{ g.p.m.}$$

Rule 2.—*At constant capacity, head is proportional to* $\frac{1}{d^{4.87}}$ (roughly $\frac{1}{d^5}$).

Example.—With a capacity of 600 g.p.m., and a diameter of 3 inches, what is the head?

The nearest figure in Table 1 is 337 feet for a 4-inch pipe, and the friction in a 3-inch pipe must be greater; hence, $h = \frac{337 \times 4.75}{1.17} = 1370$ ft.

Rule 3.—*At constant diameter, head is proportional to* $G^{1.85}$ (roughly G^2).

Example.—Same as before.

The nearest figure in Table 1 is 980 feet head for 500 g.p.m., and the friction for 600 g.p.m. must be greater; hence, $h = \frac{980 \times 600^{1.85}}{500^{1.85}} = 1370$ ft.

Rule 4.—*At constant diameter, capacity is proportional to* $h^{.54}$ (roughly $\sqrt{h}$).

Example.—Diameter = 20 inches; $h = 64$ feet per 1000 feet of length. What is the capacity?

The nearest figure in Table 2 is 44.8 feet, giving 14,000 g.p.m. The capacity will be greater; hence, $\frac{14000 \times 64^{.54}}{44.8^{.54}} = 17{,}000$ g.p.m.

CHAPTER III

CLASSIFICATION AND TYPES

34. Viewed as to the arrangement of the water end, direct-acting steam pumps may be classified as follows:

1. SINGLE SINGLE-ACTING.—This type is seldom used and must, as a rule, be equipped with large air chambers.

2. SINGLE DOUBLE-ACTING.—This type is extensively used for boiler feeding, elevator service and waterworks. Such pumps must be equipped with large air chambers and especial attention must be paid to keeping the water reasonably free from air.

3. DUPLEX DOUBLE-ACTING.—This is the most important and most extensively used type of direct-acting pumps. Duplex pumps deliver a nearly uniform flow of water and, therefore, require but small air chambers. They are used for all services.

Forming subdivisions of the above a further classification may be made, namely:

1. HORIZONTAL DOUBLE-ACTING PUMPS.
   (a) Submerged piston pumps.
   Submerged plunger and ring pumps.
   (b) Straightway piston pumps.
   Straightway plunger and ring pumps.
   Straightway inside-packed plunger pumps.
   (c) Outside center-packed pumps.
   (d) Outside end-packed pumps.

2. VERTICAL SINGLE-ACTING PUMPS.
   (a) Bucket pumps.
   (b) Plunger pumps.

3. VERTICAL DOUBLE-ACTING PUMPS.
   (a) Piston pumps.
   Plunger and ring pumps.
   (b) Outside center-packed plunger pumps.
   (c) Differential plunger pumps.

Considered from the steam end, direct-acting pumps are divided into simple, compound and triple-expansion types, according as each water cylinder is served by a single steam cylinder, a high- and low-pressure steam cylinder and a high-, intermediate- and low-pressure steam cylinder.

The above types comprise the pumps in use. Other types may possibly be found which were built to suit special conditions, but they, being exceptions, need not be included. It must be borne in mind that only those types are enumerated that are used in direct-acting steam pumps.

35. Single Pumps.—The guiding principle in a single pump with a steam-thrown valve is an auxiliary valve *d* (Figs. 18, 19 and 20), mechanically operated by the main piston *a*, and controlling an auxiliary piston *c*, which, in turn, operates the main valve *b*. The latter operates the main piston *a*, thus completing the cycle. These four elements can always be recognized in a single pump, although they may be combined in various ways or split up into several units.

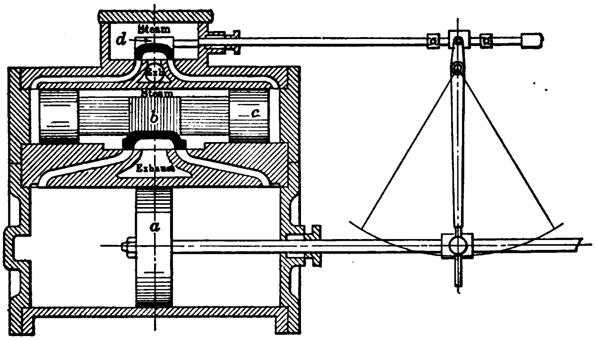


Fig. 18.—Single-valve motion, both "*D*" valves.

A point that must always be considered in designing a valve motion is, that somewhere a mechanism must be introduced which reverses the motion. If, as in Fig. 18, both valves are of the common "*D*" type, it is evident that, since each piston follows its valve, there must be a time when all four elements are at the same end of the steam cylinder. This is easily seen by moving the auxiliary valve *d* to the right, which will cause the auxiliary piston *c* to move to the right, and with it the main valve *b*, thus admitting steam to the head end of the cylinder and moving the main piston. In order to keep the pump going, the piston *a*, in moving to the right, must move the auxiliary slide valve *d* to the left; this requires a lever between *a* and *d* with a fulcrum between, so as to reverse the motion.

This result, however,.can be accomplished in a different way, namely, by employing a "*B*" main valve as in Fig. 19, and thus reversing the motion between *b* and *a*. Then when the auxiliary valve *d* moves to the right, the piston *c* and the valve *b* also move to the right, admitting steam to the right end of the cylinder and causing the main piston *a* to move to the left. In doing so, near the end of the stroke, it moves the auxiliary valve *d* to the left, starting on a new cycle. The same effect is accomplished by making the auxiliary valve a "*B*" valve and the

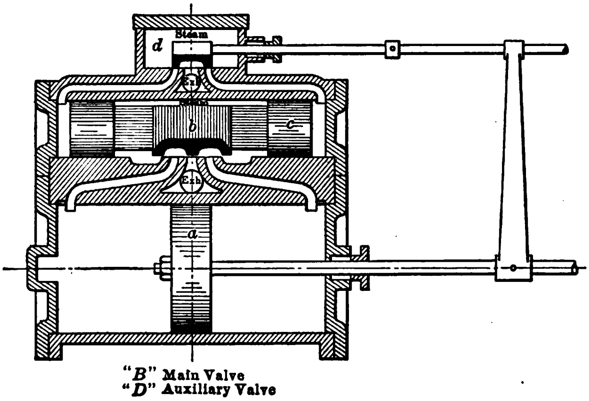


FIG. 19.—Single-valve motion, with a "*B*" main valve.

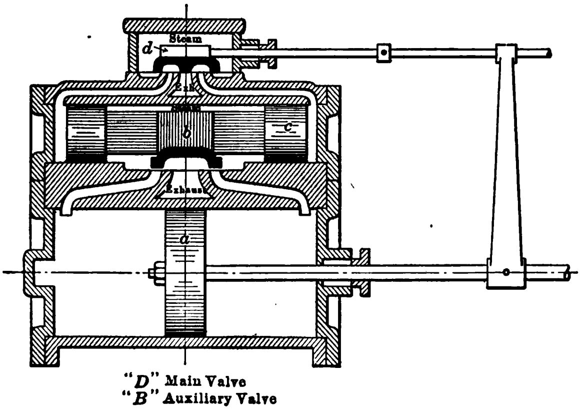


FIG. 20.—Single-valve motion, with auxiliary "*B*" valve.

main valve a "*D*" valve, as in Fig. 20. The Blake pump, in so far as the writer knows, is the only one on the market using this arrangement.

A few characteristic single-valve motions will be described, and the ingenious ways in which the different inventors have endeavored to combine some of the elements will be pointed out. They have already been divided into three groups, represented by Figs. 18, 19 and 20, according to the type of valve used. They could also be classified as to external or internal valve gear, but the proper classification is probably that according to which elements are combined.

In the following tabulation some characteristic valve motions are given:

Maker	Valve Gear	Characteristic Features
A. S. Cameron Steam Pump Wks., Phillipsburg, N. J.	Internal	Auxiliary valves are of the poppet type.
Marsh, Union Steam Pump Co., Battlecreek, Mich.	Internal	Main piston and auxiliary valve are combined. Main valve and auxiliary piston are combined.
Deane of Holyoke	External	All four elements separate.
Dean Bros. of Indianapolis	External	All four elements separate.
Laidlaw-Dunn-Gordon Vertical	External	All four elements separate.
G. F. Blake Mfg. Co., East Cambridge, Mass.	External	Auxiliary valve serves as cut-off.
M. T. Davidson, Brooklyn, N. Y.	External	Main valve and auxiliary valve are combined.
Knowles Steam Pump Works, East Cambridge, Mass.	External	Auxiliary piston and auxiliary valve are combined.
Laidlaw-Dunn-Gordon Co., Cincinnati, O.	External	Main valve and auxiliary piston are combined.

36. Cameron Pump.—One of the simplest devices is that embodied in the Cameron pump, which has no external valve gear and, therefore, no crosshead. The steam and water cylinders can be arranged quite close together and all parts of the valve gear are protected. The disadvantages are that no indicator can be applied and that the length of the stroke cannot well be measured. As a single pump, however, cannot reverse its valve without making a full stroke, this length can be assumed to be normal.

The auxiliary valve, Fig. 21, is in the form of two poppet valves *a*, one at each end of the cylinder. They are operated by the piston *b*, which, in approaching the end of the stroke, comes in contact with the valve stem protruding into the interior of the cylinder. The valve seats in a chamber, which is in communication with the exhaust through port *c*.

A poppet valve has only two significant positions, that is, it can be either open or shut, and as it seats on an exhaust port in this case, it can be either open to the exhaust or closed. A slide valve, however, has three significant positions—it may be open to the steam or closed, or open to the exhaust. This deficiency of the poppet valve has been overcome by providing a small leak into the auxiliary steam cylinder, through hole *d*, which is always open. When the piston strikes the poppet valve and opens it, the steam in cylinder *e* blows out through the large exhaust port *f*, assuming the valve piston to be to the right of central position, and the live steam cannot follow quick enough through the hole *d* to make up for what blows through into the exhaust. The result is that the steam pressure on the other side pushes the valve piston over and thus throws the main valve *g*, which reverses the piston *a*. This change is very rapid so that the loss is probably small.

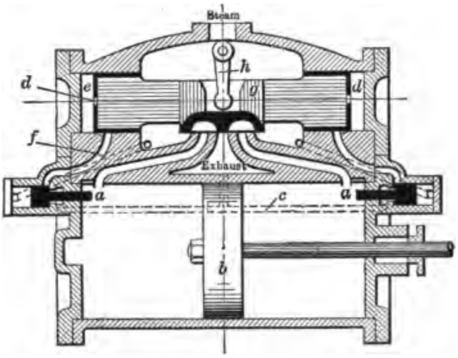


FIG. 21.—Cameron valve motion.

Single pumps, especially after they have been standing idle for some time, are apt to stick and refuse to start. Hence, a starting lever *h*, or some similar mechanism, must be provided to work the valve piston back and forth. This, of course, is necessary only in pumps where the valve motion is entirely enclosed and is not attached to any protruding valve rod by means of which it can be shifted.

37. Marsh Pump.—This is a remarkable valve motion inasmuch as the main piston and the auxiliary valve are combined into one element, also the main valve and the auxiliary piston, so as to reduce the number of elements to two; see Fig. 22. To accomplish this simplification, special means had to be employed to supply the main piston with steam. The port *a* extends downward and endwise until it reaches the head end of the cylinder, where it terminates in a little steam chest in the center of the head. To this is secured a pipe *b*, which passes through a stuffing-box and slides in a hole bored through the piston rod. From there, the steam enters the inside of the piston which is of a special construction. Small ports *c* near the ends of the main cylinder communicate with the auxiliary cylinders. In this way the steam reaches the auxiliary piston and actuates the main valve.

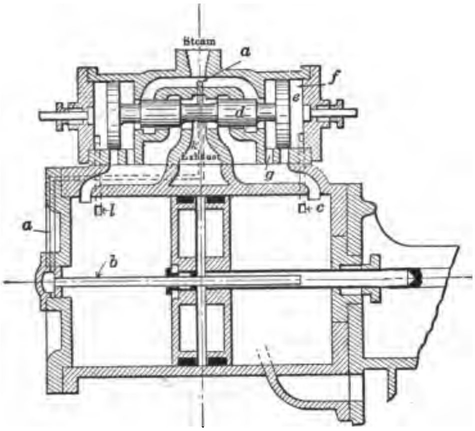


FIG. 22.—Marsh valve motion.

38. Deane Pump.—This pump, Fig. 23, has the four elements separate, with the additional feature of a cutoff, which makes it superior to the other two mentioned. The auxiliary valve *a* is in the form of a frame embracing the main valve *b*. While the main piston *c* is moving, these two valves remain at one of their extreme positions, and when the piston reaches the end of its stroke, both valves start and move until the main valve *b* has come to the center position, where all the ports are closed and the main piston stops. The auxiliary valve *a* has now opened some small ports *d* leading to the ends of the valve piston *e* which admits steam to one end and exhaust to the other; thus the main valve is shifted still further in the same direction and opens the opposite ports

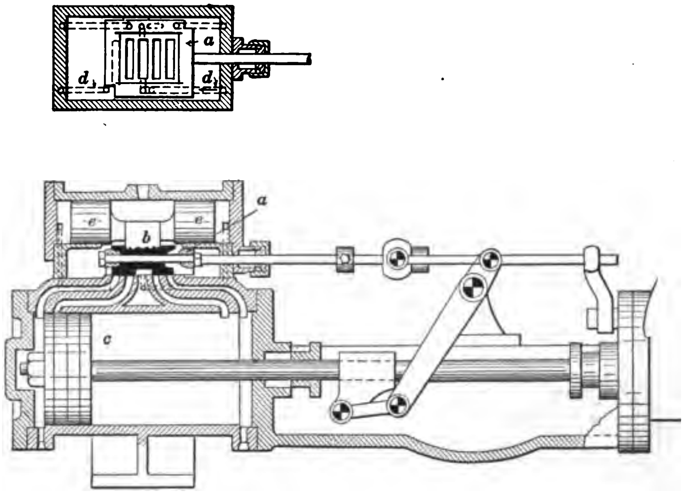


FIG. 23.—Deane valve motion.

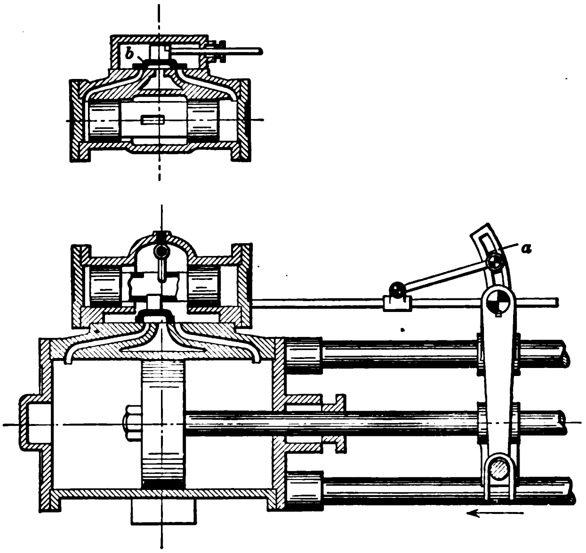


FIG. 24.—Dean Bros. valve motion.

In this pump is employed the five-ported cylinder which is a characteristic feature of the duplex pump. Instead of having one main port and one small vent port, two large ports are provided, one serving to admit the steam to the end of the cylinder and the other for exhausting the steam from the cylinder. This will later be explained more fully.

39. Dean Bros. Pump.—In this pump the four elements are also separate, but it differs from all others in this class in that there is no lost motion, each piston being rigidly attached to the valve operating the other piston. Referring to Fig. 24, the crank *a* is connected to the auxiliary valve *b* by a link-motion, whereby the travel of the valve can be changed. The valve itself is designed with liberal lap which performs the same duty as the lost motion without introducing shocks which are always more or less detrimental to high speed. This is a step in the right direction, the only point against it being the danger of having the pump stop on dead center, which is a minimum when the valves are designed without any lap.

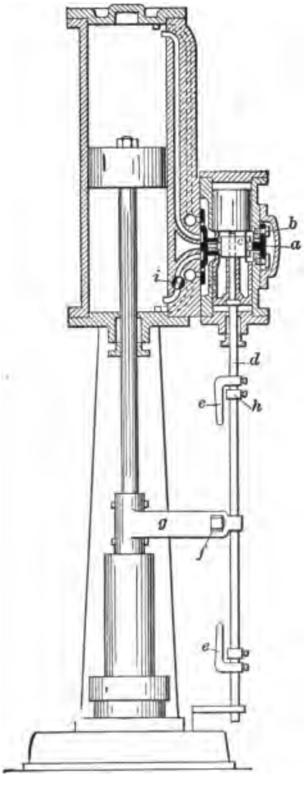


Fig. 25.—Laidlaw-Dunn-Gordon vertical valve motion.

40. Laidlaw-Dunn-Gordon Vertical Deep-well Pump.—The valve motion of this pump is designed with special consideration of the vertical position. If an auxiliary slide valve with lost motion is used in a vertical pump, during the greater part of the stroke the valve with its heavy rod is disconnected from the operating mechanism and, therefore, is apt to slide down and out of position. To avoid this, a pendulum valve *a* (Fig. 25) is employed. This is suspended from a pin *b* and has a short rack on its under side, meshing into a few teeth on the segment block *c*, which is attached to the valve rod *d*. A slight rotary motion is imparted to the valve rod *d* by the fingers *e* coming in contact with the roller *f* on the crosshead *g*. The swinging of the pendulum valve admits steam to one end of the auxiliary piston and exhausts it from the other end and thus operates the main valve.

The main valve is a "*B*" type of peculiar construction, as it is designed for a five-ported cylinder. This design was selected so as to have all parts in starting position after the pump has stood idle long enough to allow them to settle down to the lower extreme position. Should the auxiliary valve mechanism fail to act, the crosshead arm will strike one of the tappets *h* and move the main valve to the center position.

A choke valve *i* is placed in the lower exhaust port and can be adjusted so as to vary the steam forces acting on the up and down stroke, corresponding to the varying load due to the weight of the reciprocating parts.

41. Blake Pump.—Although this valve motion, Fig. 26, looks different from any of those considered, it is not difficult to recognize the four elements, namely, the main piston *a*, the main valve *b*, the auxiliary piston *c*, and the auxiliary valve *d*. The peculiarity of the arrangement lies in the fact that the auxiliary valve is interposed between the main cylinder and the main valve, forming the seat for the latter and performing the function of a cutoff. This intermediate valve *d* is moved back and forth by means of a link mechanism operating a sleeve *f*, which slides on the valve rod. This sleeve strikes the tappets *g* and moves the auxiliary valve *d*, which has small cavities or projections on the side, covering or opening small ports *e*, thus admitting steam behind the auxiliary piston and operating the main valve *b*.

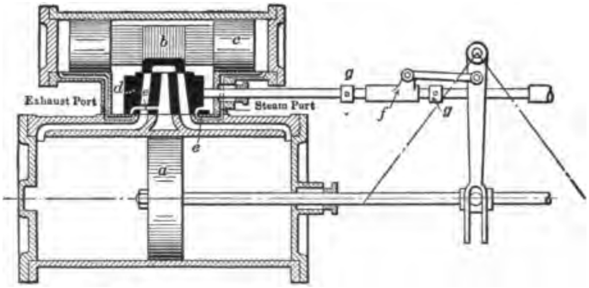


FIG. 26.—Blake valve motion.

42. Davidson Pump.—The Davidson pump represents one of the finest examples of a single-valve motion. It is positive in action and is the only pump where the main and auxiliary valves are combined. In order to let one part perform the duty of two elements, it must necessarily receive two distinct kinds of motion, a sliding motion and a rotary motion. This is accomplished by means of a cam *a*, Fig. 27, which moves a pin *b*, projecting from the main valve *c* into the cam. This cam in turning imparts a rotary motion to the main valve, placing it in position with regard to the auxiliary ports *d* which admit steam to the valve pistons. If the latter should fail to respond, then the pin will be caught in the corner of the slot *e* and will be moved lengthwise, shifting the main valve until all the ports are closed.

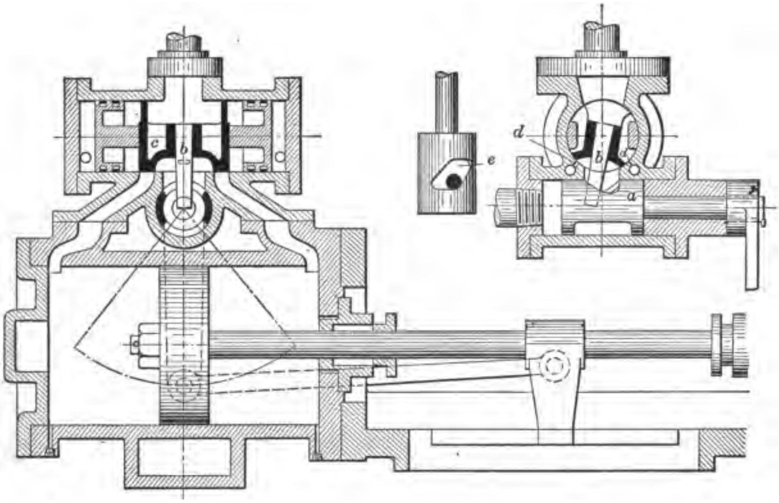


FIG. 27.—Davidson valve motion.

43. Knowles Pump.—In this pump, Fig. 28, the auxiliary piston *a* has two motions. By one, the rotary motion, it performs the duty of the auxiliary valve and by the other, the sliding motion, it operates the main valve. The crosshead is provided with an arm *b* and a roller *c* engaging the cam *d*, which is in the form of a double arm pivoted at the center. The valve rod carries an arm *e*, connected to one end of the cam *d* by a link *f*. As the main piston travels from left to right, the roller *c* strikes the arm *d*, pulling the link *f* down, and rotating the auxiliary piston. This uncovers the auxiliary ports *b* and admits steam behind the auxiliary piston which, in turn, shifts the "*B*" valve *g* to the right, causing the main piston *h* to move to the left.

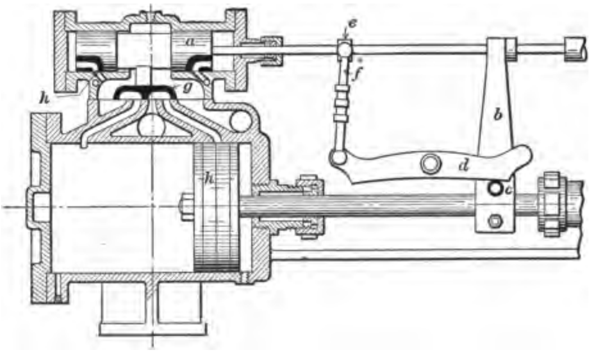


FIG. 28.—Knowles valve motion.

44. Laidlaw-Dunn-Gordon Pump.—This design, Fig. 29, is commendable for its extreme simplicity. The main valve and the auxiliary piston are in one piece, and the auxiliary valve is a small ordinary slide valve working over ports on the side of the main steam chest. No cutoff is provided, the action of the main valve being dependent upon the proper operation of the auxiliary valve. To prevent the piston from striking the head the steam ports are arranged to produce a cushion. The port enters the cylinder a short distance from the head and the piston, in approaching the head, covers the port and the entrapped steam forms a cushion. A small cushion port lets the entrapped steam escape and on reversal admits steam behind the piston.

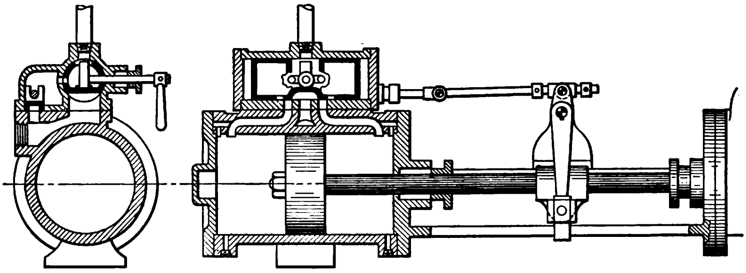


FIG. 29.—Laidlaw-Dunn-Gordon horizontal valve motion.

45. Auxiliary Pistons.—In determining the size of the auxiliary piston one must be governed largely by experience. It could, of course, be calculated from the force required to move the slide valve, by multiplying the unbalanced area by the steam pressure and by the coefficient of friction. To this must be added the friction of the valve piston, and if this sum be divided by the steam pressure, the result is the area required in the valve piston. However, the entire steam pressure is not available for moving this valve, for a pump must work with high steam pressure as well as with low steam pressure, and it must work when it is in first-class condition and when it is not. A practical rule is, to make the diameter of the auxiliary piston 45 per cent. of the diameter of the main piston.

46. Defects of Single Pumps.—Single pumps, such as described, are poor things to handle an unyielding substance such as water. Although there is no appreciable pause at the end of the stroke and the reversal is instantaneous, yet there is, of necessity, a momentary stoppage which causes an interruption in the flow. To remedy this, large air chambers are usually employed which are supposed to take up a certain amount of water during the stroke, and replenish it during the pause. An air chamber, however, if it is an essential part of the pump, is not very reliable, as it requires constant attention.

Water under atmospheric conditions contains about 2 per cent. air in solution, and in passing through the pump, this air is compressed and occupies less space than originally. If the air molecules are considered as filling the interstices in the water, it is evident that, after compression, additional room is made available for more air. The air in the air chamber will now enter and fill the void spaces; that is, the water has absorbed the air. The higher the pressure, the quicker this absorption takes place, so that it is not at all practical to use an air chamber for pressures over 300 pounds per square inch.

Putting an extra supply of air into the chamber, by opening a snifting valve in the suction pipe, is not recommended. One of the first rules to follow in designing a pump is to avoid all air pockets; that is, the interior of the pump must be such that the water is always rising and there must be no pockets where air can accumulate. Yet in the best of pumps, where this rule has been strictly followed, the presence of air is sometimes detected. It is usually manifest by a short jump of the plunger at the beginning of the stroke.

The air, in common with the other permanent gases, has the property of adhering to the surface of a solid so firmly that it cannot be removed except by heating or by brushing it off. This film of air is constantly compressed and re-expands at every stroke, and is often the cause of severe irregularities in the action of direct-acting pumps.

Pumps with compulsory motion, such as the crank-and-flywheel types, or power pumps, are not affected by air, in so far as outside appearances go. The air may be present and may seriously affect the displacement and the capacity of the pump, but there is no sign of it in the operation. The presence of air can be detected by means of the indicator which will show compression at the beginning of the discharge stroke and expansion at the beginning of the suction stroke. Fig. 30 is a diagram taken from a pump containing air; *AB* is the compression curve, *CD* the expansion curve, S the stroke, which is proportional to the displacement of the plunger, and S_1 the amount of water actually pumped. Then $S - S_1$ represents the loss in capacity due to the presence of air.

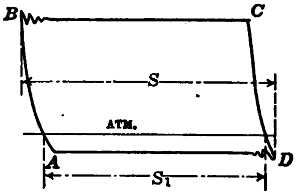


FIG. 30.—Diagram from pump body containing air.

Therefore, unless some means is at hand to keep the air chamber supplied with air, a single pump will frequently work with shocks and produce water-hammer in the pipes, which may be so severe as to cause breakage.

Single pumps are excellent for removing the condensate and air from surface or jet condensers, for which service they are called "wet vacuum pumps." Here the amount of air is so large (about 50 per cent.) that it forms a highly elastic cushion and thus mitigates the shocks. Moreover, as a single pump is bound to make a full stroke, since it cannot reverse until the piston has come to the end of the stroke, it is very efficient for removing the air; much more so than a duplex pump which is liable to short stroke. The necessity of a uniform discharge does not apply in this case because the pump handles only a small amount of water. For this purpose, a duplex pump would be more costly and throw an additional burden on the attendant, who will have to take care of double the number of stuffing-boxes and pistons without getting any adequate return for this complication.

47. Duplex Pumps.—If the design of a duplex pump is analyzed the same elements are found as in the single pumps, namely, two pistons and two valves operating in a cycle. The difference is that both pistons do their share of the work, there being two main pistons and two main valves, instead of one main piston, one main valve, one auxiliary piston, and one auxiliary valve. As already shown in the single pump, the reversal can be produced by making one of the valves a "*B*" type, and it would, of course, be possible to design a duplex pump on this principle. In fact, the first duplex pumps were so equipped and the valves were driven by tappets. One report has it that two relief-valve-motion pumps had been installed end to end in the St. Nicholas Hotel in New York, and in watching the motion of the pistons and valves, it was noted that the piston of one pump and the valve of the other were traveling simultaneously and that they could be attached and made to travel together.

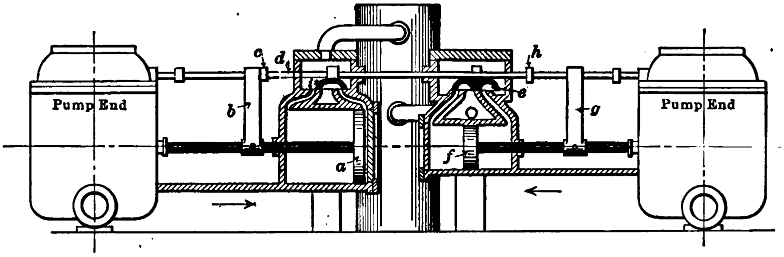


Fig. 31.—Cross-compound with duplex valve motion (1871).

The relief-valve motion at that time was not a perfect mechanism, as there was nothing to stop the motion of the piston, the cushion not yet having been invented. Moreover, the opening of the relief port, while a good feature at the end of the stroke, did not prove so great a benefit after reversal and at the beginning of the stroke, when there was no resistance until the relief port had closed.

The suggested remedy was tried and proved a great success, the reversal being accomplished by making one valve of the "*D*" and the other of the "*B*" type, as in Fig. 31. This represents a compound pump, a type which was developed later, but it will serve to illustrate a duplex pump if the two cylinders are considered the same size. The piston *a* of the left-hand pump has just completed its outboard stroke and its arm *b* has struck the tappet *c*, attached to the valve rod *d* of the right-hand pump. It has thereby moved the "*B*" valve *e* of the right-hand pump to the right, admitting steam to the right of the piston *f*, and moving it to the left. At the end of its stroke, the arm *g* will strike tappet *h* and move the "*D*" valve *i* to the left, causing the piston *a* to follow and also travel to the left.

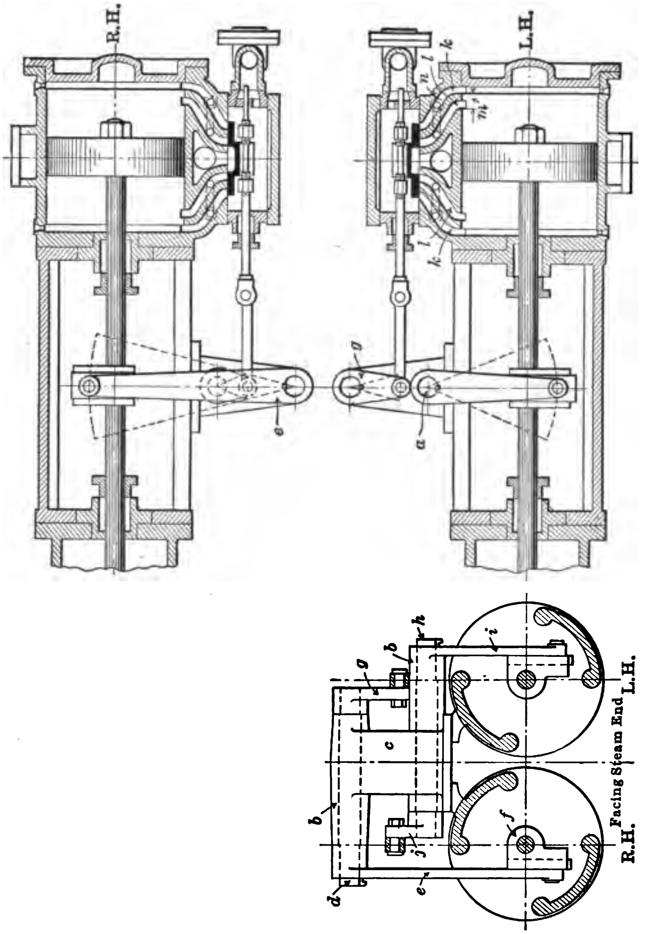


Fig. 32.—Duplex valve motion, all parts shown in center position.

The advantage of having both sides alike, however, was so obvious that means were sought to effect this reversal otherwise, and later pumps were equipped with two "*D*" valves, one driven by a direct-motion mechanism and the other by a reverse-motion mechanism.

Before going any further it will be necessary to explain what has been adopted by pump manufacturers as *right hand* and *left hand*. Standing at the steam end and facing the water end, the side on the left is the *left-hand side* (*LH*), and the one on the right is the *right-hand side* (*RH*).

48. Duplex Valve Motion.—Recalling that with a "*D*" valve the piston follows its valve, the four elements of a duplex pump will be seen to move in the following cycle: Assume both pistons to stand in a central position, then

1. *RH* valve moves inboard.
2. *RH* piston moves inboard.
3. *LH* valve moves inboard.
4. *LH* piston moves inboard.
5. *RH* valve moves outboard.
6. *RH* piston moves outboard.
7. *LH* valve moves outboard.
8. *LH* piston moves outboard.

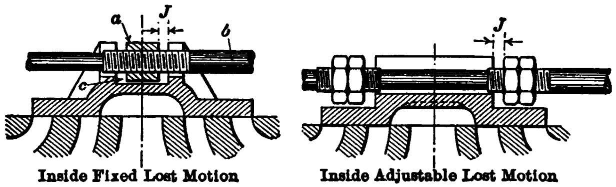
This cycle resolves into two groups: one in which all the parts move inboard, and the other in which all the parts move outboard. Between two such groups the direction of motion must be reversed, that is, between (4) and (5) the left-hand piston, in moving inboard, must move the right-hand valve outboard. This is accomplished by placing the fulcrum *a* (Fig. 32) of the arm between the piston rod and the valve rod.

The motion is transmitted from the pistons to the valves by two rock-shafts mounted in bearings *b*, formed in a single piece *c* called the "cross stand." To the right-hand end of the upper rock-shaft *d* is fitted a long arm *e*, the lower end of which is attached to the crosshead *f* on the piston rod. To the left-hand end of the rock-shaft *d* is fitted a short crank *g*, attached to the valve rod by means of a clevis and link.

To the left-hand end of the lower rock-shaft *h* is fitted a long arm *i* similar to that on the right side but shorter by the distance between the rock-shafts. To the right-hand end of the lower rock-shaft is fitted a crank *j* similar to the one on the left-hand side but shorter and pointing up instead of down. This combination of long arm and rock-shaft and crank constitutes the mechanism which reverses the motion.

49. Lost Motion.—The study of the relief-valve-motion pump revealed the importance of a certain pause in the flow of water at the end of the stroke which gave to the valves an opportunity to seat quietly. Although the piston did not pause, yet the opening of the by-pass between the two ends of the pump body stopped the flow of the water and gave the same effect. To embody this feature in the duplex pump, the lost motion or valve rod clearance was invented. This clearance can be had in various ways.

Fig. 33*a* shows how the clearance is obtained in small pumps. A valve rod nut *a* is placed on the rod *b*, the nut being thinner than the space *c* in the slide valve. Here the lost motion is fixed and cannot be changed except by inserting a nut of different length.

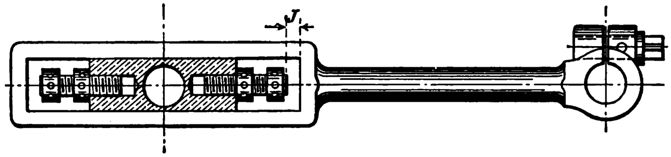


Inside Fixed Lost Motion

FIG. 33*a*

Inside Adjustable Lost Motion

FIG. 33*b*.



Outside Adjustable Lost Motion

FIG. 33*c*.

FIGS. 33*a*, 33*b* and 33*c*.—Lost-motion devices.

In Fig. 33*b* this lost motion is adjustable, although the pump must first be stopped and the steam chest cover removed. This is to prevent anyone from interfering with this adjustment, as such pumps are set for the conditions under which they are to run and it is not desirable that any adjustments be attempted.

On large pumping engines that are operated by skilled attendants, the clearance is provided on the outside in a lost-motion link (Fig. 33*c*), which can easily be adjusted while the engine is running. Such pumps are built for economy and it is well to have some convenient means at hand for operating them at the most economical point.

Fig. 34 represents graphically the action of a duplex pump, the paths of the two pistons being represented by the two vertical lines. The *LH* piston is supposed to stand at *J* and the *RH* piston at *C*. The connecting lines of the various points have no meaning other than to show which points belong together. Assuming that the pistons travel with uniform velocity throughout the stroke, it will be seen that the *LH* piston travels from *J* to *I* while the *RH* piston travels from *C* to *A*, finishing its stroke, but the valve governing the *RH* piston is operated by the *LH* piston and does not open until the *LH* piston has arrived at *H*. If there were no lost motion, it is clear that the *RH* valve would be in its center position which is the position of reversal, when the *LH* piston has reached the center of its stroke, but owing to the valve rod clearance the motion of this valve is retarded and it does not attain its center position until the *LH* piston has

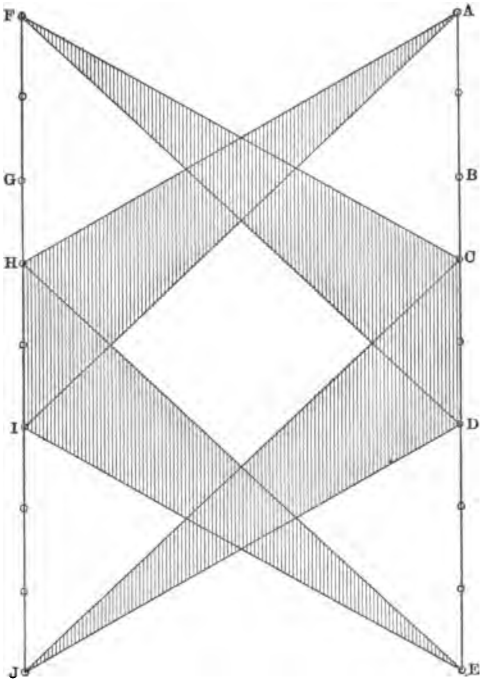


FIG. 34.—Motion diagram of duplex pump piston.

arrived at *H*. Therefore, while the *LH* piston traveled from *I* to *H*, the *RH* piston must have paused at *A* during a period equal to one-fourth of the stroke. This pause is indicated by the two diverging lines *AI* and *AH*. The two pistons are now in the same position relative to each other as before and will go through the same movements.

In Figs. 35 and 36 this diagram is in part repeated and the valve motion, as well as the steam cylinders with their valves, are

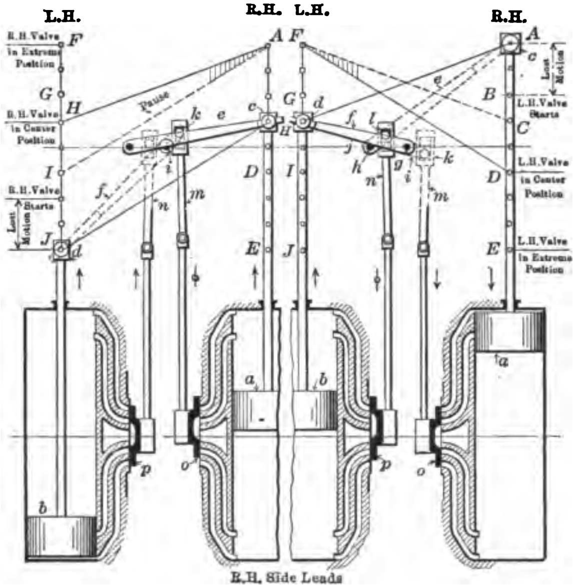


FIG. 35. FIG. 36.

FIGS. 35 and 36.—Duplex valve motion.

added, all parts being shown in two positions. The steam pistons *a* and *b* are connected to the water pistons by rods to which are attached suitable crossheads *c* and *d* for operating the valve motion. These crossheads engage the ends of the valve motion arms *e* and *f* by links, sliding blocks, or swivels, giving to these arms an oscillatory or rocking motion. The arms are keyed to shafts *g* and *h* extending across the machine, and short arms *i* and *j* are keyed to the other ends of the rock-shafts.

In Fig. 36 the *RH* piston is just starting on the downward stroke, and the *LH* piston is at a point five-eighths from the beginning of the stroke, traveling upward with maximum velocity.

Assume, for simplicity, that the pistons move with uniform velocity throughout the stroke and that each, at the end of its stroke, pauses a length of time equivalent to a travel of one-fourth the stroke. Then by the time the *LH* piston has reached the end of the stroke, the *RH* piston will have traveled an equal distance and its crosshead will have come to point *C*. In doing so, the lost-motion block *l* will be moved downward, and when the *RH* crosshead reaches point *B*, the block *l* will come in contact with the link *n* and begin to move the valve *p* downward from the extreme position shown. From now on, the ports of the *LH* piston are gradually closed until, when the crosshead of the *RH* piston has reached point *D*, the valve *p* has reached its central position and closed all the ports. But since it was assumed that the crosshead of the *LH* piston arrives at *F* when that of the *RH* piston is at point *C*, the *LH* piston cannot reverse and must rest while the *RH* crosshead is traveling from *C* to *D*, which is the time during which the valve is shifted from the position shown to the central position. The *LH* valve then begins to open and reversal of the *LH* piston takes place, until when the *RH* piston has come to the end of its downward stroke, with its crosshead at *E*, the *LH* valve is full open.

A study of this diagram furnishes a rule for the adjustment of the lost motion. However, it is well first to consider which side leads. For a long time, it was the general practice to design all valve motions with the *RH* side leading. This is now not strictly adhered to, which is to be regretted as any attempt at adjusting requires first a study of the valve motion. The side having the long rocker arm is the leading side. Where the two are of equal length, that which has a crank attached, pointing in the same direction, denotes the leading side.

Assuming that the *RH* side leads, if the *LH* piston short-strokes (stops before its crosshead has reached *F*), the lost motion must be increased at *k*. This will have the effect of making the *LH* crosshead travel farther than *H*, say to point *G*, before the *RH* valve passes its center position, and thus starts the *RH* piston. Then the *LH* crosshead has only a short distance to travel, from *G* to *F*, while the *RH* crosshead travels from *A* to *D*, thus giving the *LH* piston more time to finish its stroke.

If the lost motion is made such as in Fig. 33*b*, it is evident that the adjustment must be made on the opposite end from the one that must be adjusted if a lost-motion link like that in Fig. 33*c* is used. However, a rule can be formulated that will hold good in either case.

General Rule for Adjusting Lost Motion

With either side leading, to lengthen the stroke on either corner of the pump increase the lost motion of the valve on the opposite side, and at that end of the lost motion block which is in contact with the link end.

The amount of lost motion depends upon so many conditions that it cannot be determined beforehand. A general rule is to make it equal one-third the travel of the crank pin at contact stroke, and then make the proper adjustment when the engine is running under contract conditions.

In doing so, it should be borne in mind that the more the lost motion, the longer will be the pause at the end of the stroke. This loss of time must be made up by an increase in the actual piston speed, the average or mean piston speed being fixed by the capacity of the pump.

If a pump will run quietly, making a full stroke without any lost motion, then the plunger will reverse instantly and no time will be lost.

Going to the extreme, as did the Worthington Co. when they attempted to run their duplex air pumps with the stroke of the crank-pin nearly all lost motion, that is, when they used a tappet valve motion, then one piston will travel right up to the end of the stroke, move the valve on the opposite side, and pause until the other piston has reached its end of the stroke and has moved the valve. The action of the pump is then not any different from that of a single pump, and with the disadvantage of having all the parts double and of having one pump standing idle while the other is doing the work. One single pump of slightly larger size would have been much more appropriate and, at present, the use of duplex air pumps has been abandoned entirely.

50. Cushion.—The third difficulty arose when it was found that there was nothing to stop the motion of the piston at the end of the stroke. The two sides acting independently, made it impossible to time their action so that each valve would shift just before the piston reached the end of the stroke; therefore, some other means had to be devised to accomplish this purpose.

This was found in designing the steam cylinder with two sets of ports, the outer port acting as the steam admission port, and terminating at the end of the cylinder so as to guide the incoming steam into the clearance space behind the piston; the other acted as the exhaust port and terminated a short distance from the head. The piston, in approaching the head, covers this exhaust port and entraps the steam in the space between the piston and the head, compresses it by force of its momentum and thus comes to a stop. Thus originated the five-ported cylinder, a characteristic feature of the duplex pump.

This is shown in Fig. 32, the outer ports *k* being the steam ports and the inner ports *l* the exhaust ports. The distance *m* from the outer edge of the exhaust port to the head forms the cushion. Unfortunately, there is no convenient means of changing this distance to adjust the cushion; therefore, on the larger pumps, over 10-inch stroke, a passage is provided between the ports *k* and *l* and a valve *n* is inserted, which can be opened should the piston cushion too heavily. The port *k* is for steam admission only, hence it can be made a little smaller than the exhaust port *l*.

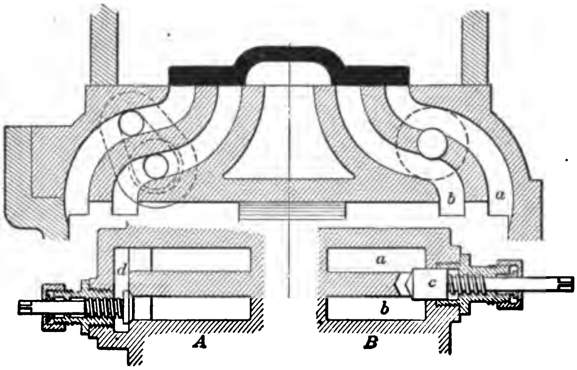


Fig. 37.—Cushion valves.

Figs. 37, A and B show different types of cushion valves. In the latter, a hole is drilled lengthwise into the bridge between the two ports *a* and *b*, and this hole, being larger in diameter than the width of the bridge, will open up communication between the two ports. A solid plug *c* is then inserted, means being provided to screw it in and out. When all in, it is supposed to fill the entire hole and close off communication between the two ports. It is obvious that, unless a driving fit is made, the passage can be closed only in a very imperfect manner. A better plan is shown at *A*, where a separate port *d* is cast on, connecting the two main ports and a regular seated valve, which can be ground in and made tight, is used to close off communication.

It is evident that the five-ported cylinder, shown in Fig. 32, presents considerable waste room or clearance. Although most of the slide valves are designed after this fashion, there is really no necessity for it. Much of the clearance could be saved by making only one main port as at *a*, Fig. 38, which serves as both steam and exhaust port, and terminates a short distance from the head to form the cushion. Then a small auxiliary port *b* can be added, terminating close to the head. The latter will then admit steam behind the piston for starting, after which port *a* will be uncovered and the full amount of steam admitted. In this case the slide valve must be designed so as to open port *a* for steam or exhaust and port *b* for steam only. This is accomplished by making the cavity *e* only as wide as port *a*, while the width of the whole valve must be sufficient to cover both ports. As seen in the illustration, the port *b* is always ready for steam admission whether the cushion valve *d* is open or closed. The area of the cushion valve opening should be 0.5 per cent. of the area of the piston.

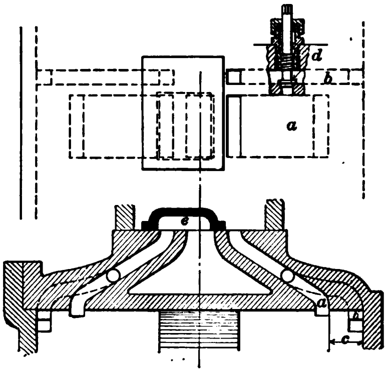


Fig. 38.—Slide valve with small steam port.

In the following table, two sets of strokes are given—*nominal stroke* and the *contact stroke*. It would be dangerous to run a pump the full length of the stroke, in other words, to let the piston travel close to the head, as a slight change in conditions might cause the piston to strike the head. Therefore it is customary to allow a certain clearance between the piston and the head. This clearance is, however, only an assumed quantity and not fixed, as in a steam engine. If a pump is designed for a stroke of 12 inches, then ½ inch is added to the stroke at each end, and the total stroke of 13 inches is called the *contact or bumping stroke*, and the former stroke of 12 inches, the *nominal stroke.* The pump is adjusted for this nominal stroke.

The following table gives good proportion for the length of the cushion (*m*, Fig. 32):

Table 6

Nominal stroke, inches	Contact stroke, inches	Cushion, inches
3	3-½	⅝
4	4-½	¾
5	5-¾	⅞
6	6-¾	1-⅛
8	8-¾	1-¼
10	11	1-½
12	13	1-¾
15	16	2
18	19-½	2-¼
24	26	2-¾
30	32	3-⅛
36	38	3-½

51. Extra Long Cushion.—On low-service pumps, which include pressure pumps for comparatively low pressures, where the weight of the reciprocating parts is heavy, it is often necessary to provide an extra long cushion in order to stop the momentum of the moving masses. For this purpose an extra long cylinder with an extra wide piston is sometimes employed. If the cylinder cannot be made longer than the regular one, as, for instance, when the pump is built and it is found to require a long cushion, but the new cylinder must fit in the place of the old one, the new cylinder can be made with a long cushion and the piston with a ring which is as wide as its face. In this way a cushion is obtained as long as the face of the piston will permit; this is illustrated in Fig. 39. Further on, under the "Design of Compound Pumps," the conditions will be stated under which it is advisable to provide extra cushioning effect.

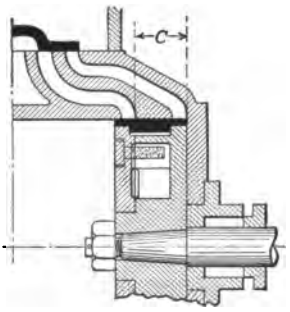


Fig. 39.—Extra long cushion.

52. Small Submerged Piston Pumps.—Small pumps are built on the plan of the submerged piston pump shown in Fig. 40. This may be fitted with a driven lining, in which case the pump body is bored out and a brass tube is forced in. If fitted with a packed piston, the tube need not be bored out after forcing in, but if the piston is solid or if it is fitted with metallic packing rings it is necessary that the bore should be absolutely true and in this case the tube must be bored out.

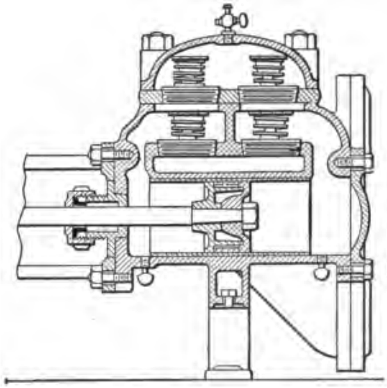


Fig. 40.—Small submerged piston pump with driven lining.

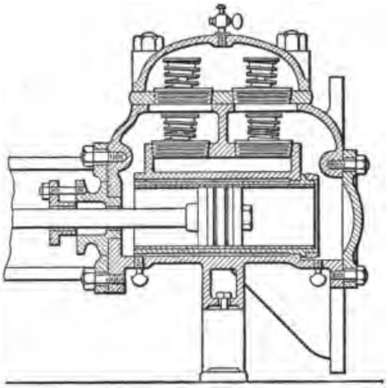


Fig. 41.—Small submerged piston pump with removable lining.

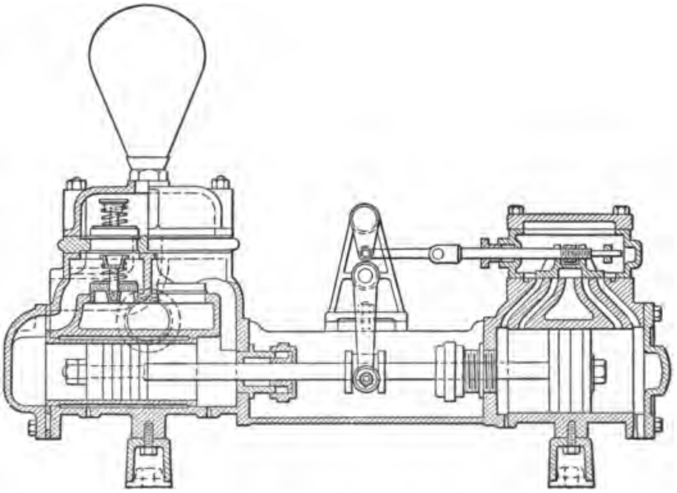


Fig. 42.—Submerged piston pump with flanged removable lining.

Again, the pump may be fitted with a removable lining, which is a bronze casting provided with a flange by which it is fastened to the pump body. Sometimes, where this flange is small, it is gripped by a binder of cheaper material, such as malleable iron, which binder is bolted on; see Fig. 41. In Fig. 42 the lining is formed with a large flange which is bolted in between the pump body and the head.

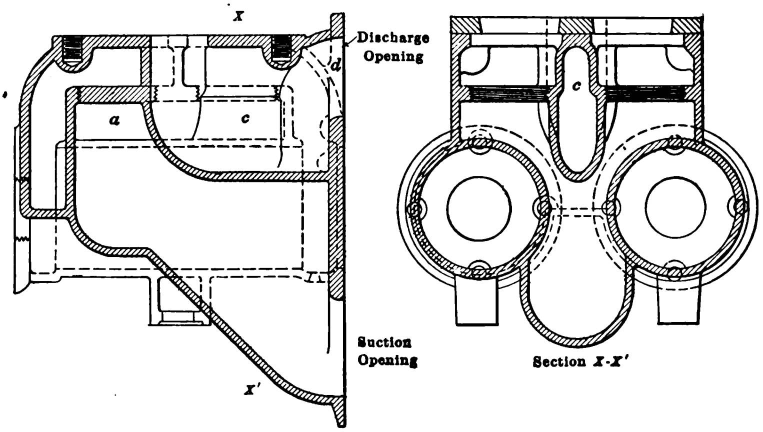


FIG. 43.—Sections of pump with end suction.

This type of submerged piston pump was developed because on such small pumps there is no space for handholes. The discharge valve deck is, therefore, made separate so that it can be removed, and in this way access is gained to the suction valves which are located in the suction valve deck cast integral with the pump body.

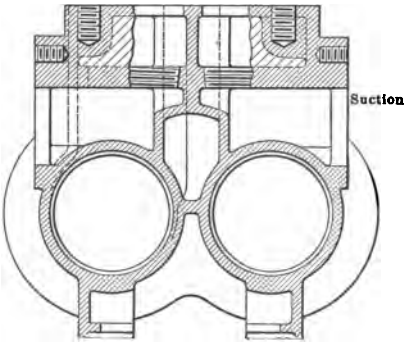


FIG. 44.—Cross-section of pump with side suctions.

The suction valve deck is located so as to leave a space "*S*" between it and the pump barrel (see Fig. 48), which space forms part of the suction chamber. If the distance between centers is sufficient, this chamber extends down between the pump barrels and terminates in a round hole at the end of the pump. Fig. 43 shows sections of a pump with end suction, while Fig. 44 represents a section of a pump where the distance between centers is too small for this construction and two side suctions are provided. One of the openings is blanked off, and the connection can be made on either side. Figs. 45 and 46 are outside views of these two types.

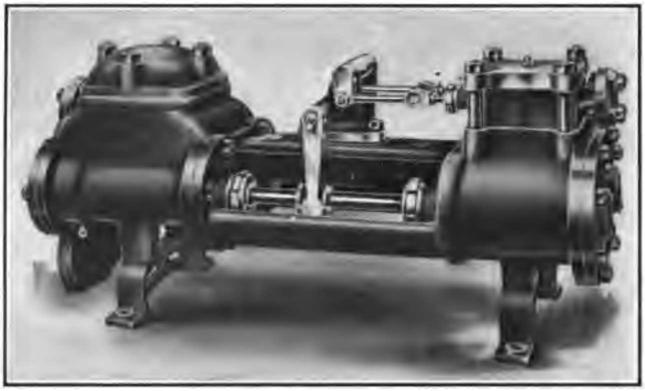


FIG. 45.—Outside view of pump with end suction.

The same type can be made as a plunger and ring pump, which is fitted with a long barrel working in a short ring, while a piston pump is fitted with a short piston working in a long sleeve. A plunger and ring pump of the submerged type is illustrated in Fig. 47. The ring is held in place by a binder of malleable iron and the back head of the pump forms a hood to provide space for

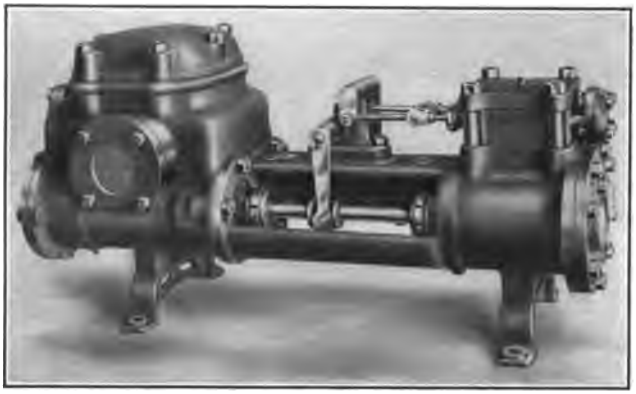


FIG. 46.—Outside view of pump with side suctions.

the plunger, otherwise the design is the same as the piston pump with removable lining.

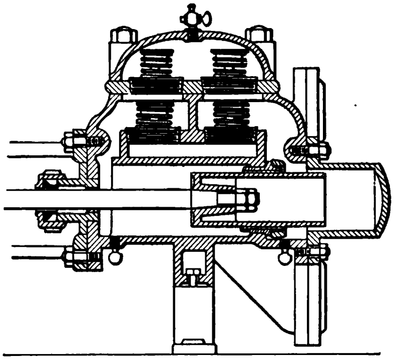


FIG. 47.—Small submerged plunger and ring pump.

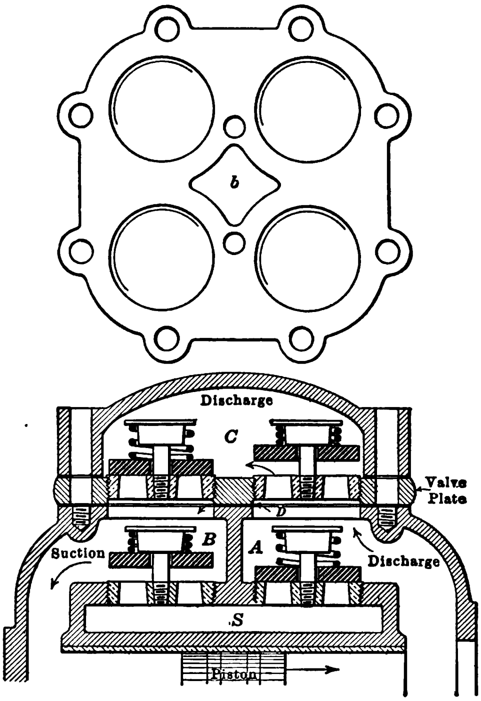


FIG. 48.—Valve plate with extra bolting.

The discharge valve deck is covered by a valve plate containing the discharge valve. This plate has a hole *b* (Fig. 48), either at the center or at the end, communicating with a curved port *c* (Fig. 43), which likewise terminates at the end of the pump in the discharge opening *d*. Thus, it is possible to remove the cap without breaking any pipe connections. Sometimes the cap is fitted with an air chamber, but small pumps run better without this addition. The pulsations of the water level in the air chamber right over the valves seem to have a disquieting effect on their operation.

On pumps with pistons 4½ inches and over, the discharge valve plate becomes so large that it must be provided with bolts in the center for the purpose of holding the gasket in place; Fig. 48 shows the bolt holes. The necessity for this will be evident by examining the three chambers *A*, *B* and *C*, the first two being separated by a joint *D*. When the pump is in operation, there is pressure in two of the chambers, say *A* and *C*, but not in *B*, which is subjected to suction. Between *A* and *C*, the valve plate is balanced, but between *B* and *C* the pressure is downward and, at no time, can the valve plate lift away from its seat. But the gasket at *D* is subjected to a pressure sideways from *A* toward *B*, and unless the valve plate is held down firmly, this gasket is liable to blow out.

On very small pumps all four cylinders, namely, the two steam cylinders and the two pump bodies, are cast in one piece. On pumps 4½ × 2¾ × 4 and over, it is preferable to separate the steam and water ends so as to have only one steam end pattern which can be combined with any type of water end. This also makes the repairs less costly should one of the ends prove defective. The two steam cylinders, with their yokes or connecting pieces, called cradles by some manufacturers, always form a single casting. In this way no pipe connections are required, much handling and machine-work is saved, and the distance between centers can be kept small.

A good rule for the distance between centers of the water cylinders is to make it 1 inch more than the diameter of the steam cylinder. This gives a broad joint between the two bores.

These small submerged piston pumps are usually built for two pressures, namely, 150 pounds per square inch for general service and 75 pounds for tank or other low service. In the former, the ratio of cylinders is about 2½ to 1, while in the latter about 1½ to 1. The largest size that can be built of this design, practically, is 10 × 6 × 10-inch.

53. Large Submerged Piston Pumps.—If the dimensions of the pump allow of handholes being provided for access to the valves, designs similar to those in Figs. 49 to 53 may be used. Fig. 49 shows a pump similar to that of Fig. 43, except that the whole body is a single casting. The discharge valve plate and the force chamber are cast integral with the pump barrel and a cover plate gives access to the discharge valves. The suction valves are accessible through the handhole on the side, as shown in Fig. 50. The former illustrates a pump with end suction and the latter, one with side suction. The valve chamber is circular, hence the designation "turret pattern."

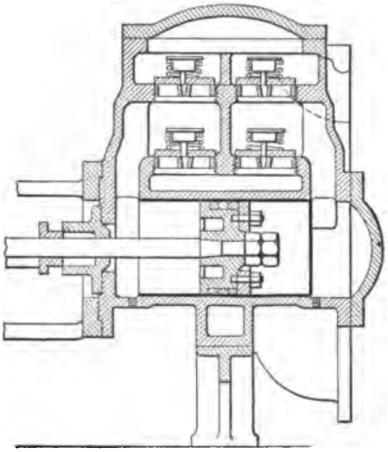


Fig. 49.—Submerged piston pump with handholes.

Figs. 51 and 52 show submerged piston pumps with rectangular valve chambers and the discharge on top. A third handhole is provided on the side for access to the discharge valves. The defect of this design is the reversal of the water in the ports, which takes place at every stroke. The water must be brought to a full stop and must then be set in motion again in the opposite direction. In small pumps the inertia of the water is too small to do any harm, but in large pumps it may cause such disturbances that the pump

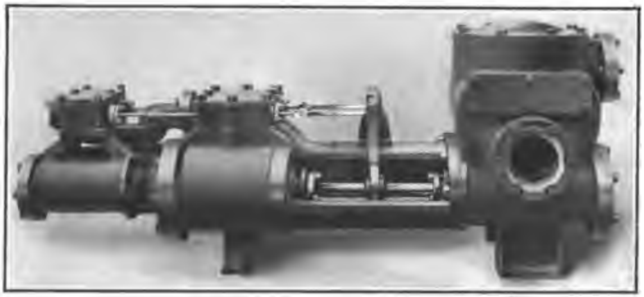


Fig. 50.—Submerged piston pump (turret pattern) with handholes.

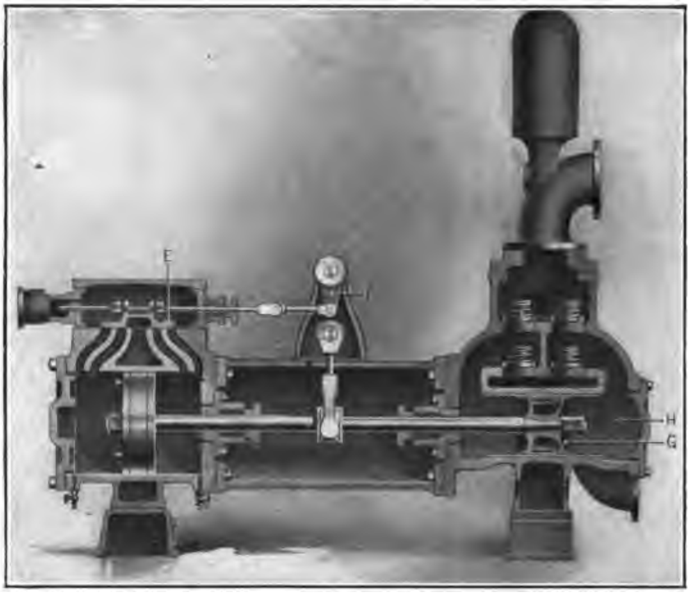


FIG. 51.—Submerged piston pump with discharge elbow.

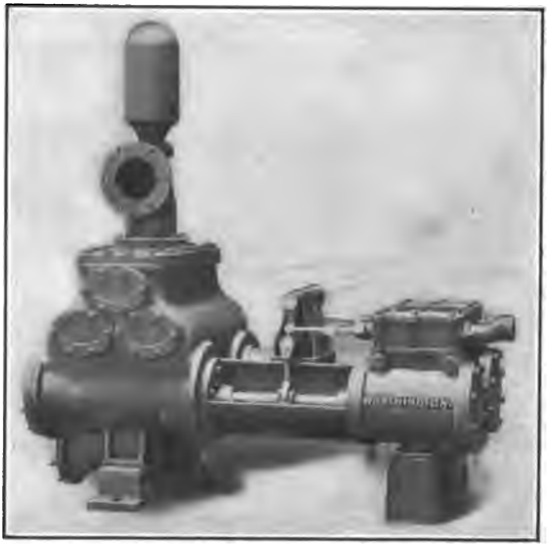


FIG. 52.—Outside view of submerged piston pump with discharge elbow.

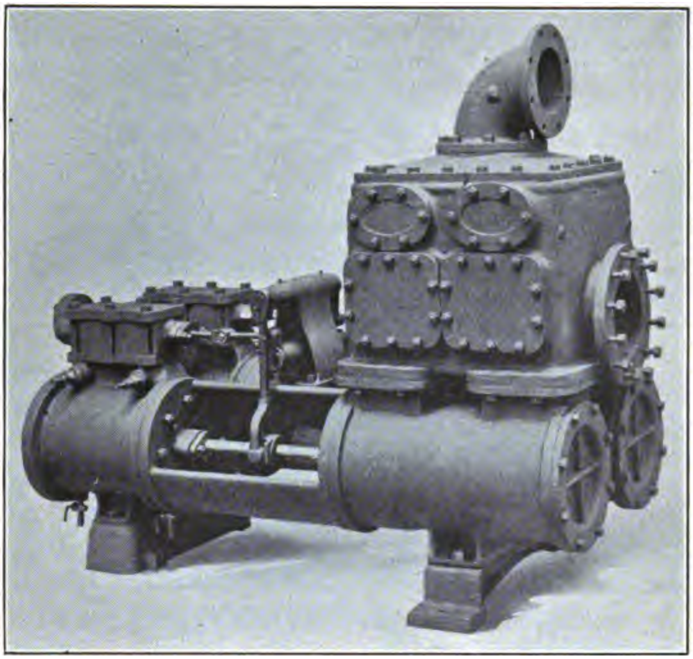


FIG. 53.—Sectionalized submerged piston pump with common valve chamber.

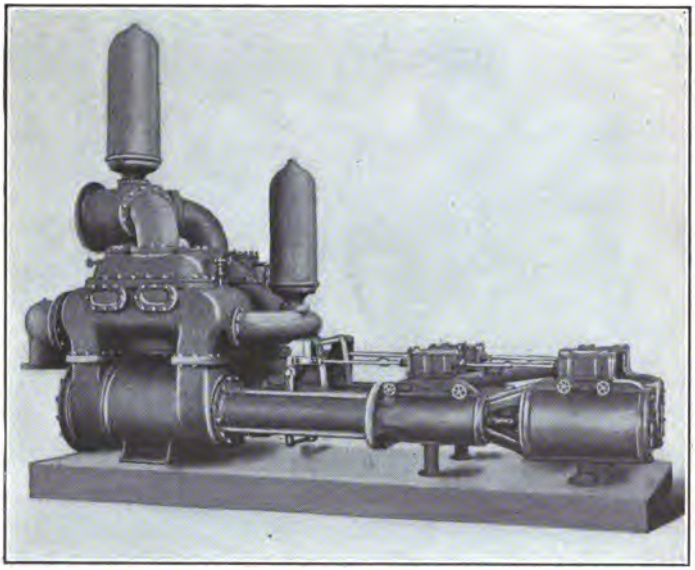


FIG. 54.—Submerged piston pump with separate valve chamber.

can run at a moderate speed only. Such pumps are built up to 14 inches and 150 pounds for general service, and up to 22 inches and 75 pounds for low service. These large pumps are sectionalized as shown by Fig. 53, the two pump barrels being separate castings with a valve chamber mounted over them.

Fig. 54 shows a large low-service submerged piston pump where two valve chambers are side by side. Sometimes they are fitted with outside packed plungers.

The submerged type of pump is recommended for handling hot water and for condenser service. The piston is always surrounded by cool and de-aerated water which protects the packing and increases the suction action of the pump.

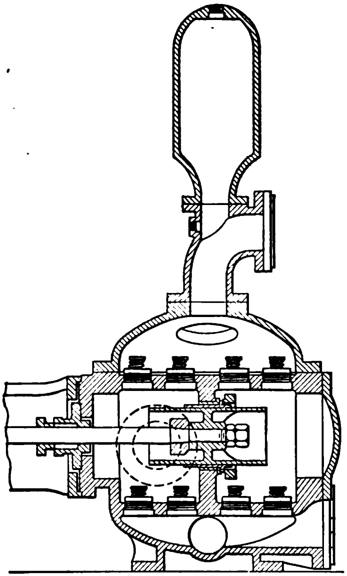


Fig. 55.—Straightway plunger and ring pump.

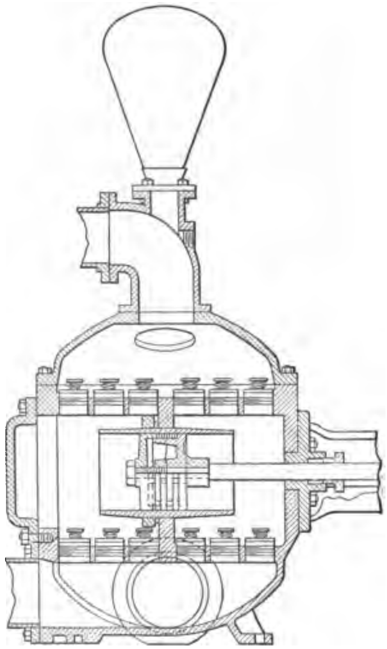


Fig. 56.—Straightway piston and sleeve pump.

54. Straightway Pumps.—For conditions where a more expensive and better design is warranted, the type designated as the straightway pattern is recommended. Here the water flows through the pump in a practically unchanged direction and while its direction of flow is not absolutely straight, yet it is more nearly straight than in the types previously described. When the plunger comes to a stop at the end of the stroke the water does not necessarily stop at the same instant, but, owing to its inertia, may continue to flow through the pump for a short time. In the submerged piston pattern the piston in reversing must overcome this tendency, bring the water to a full stop, and reverse its direction.

This type may be built as plunger and ring, or piston and sleeve pumps. The former has a long plain plunger working in a short ring, and the latter has a piston usually packed with fibrous packing working in a long sleeve. Figs. 55 and 56 illustrate respectively sections through the water ends of the plunger and ring, and the piston and sleeve types.

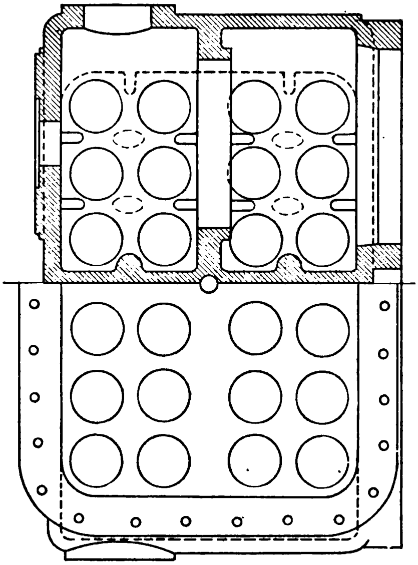


FIG. 57.—Plan of straightway pump body.

In the smaller sizes, from 6 to 12 inches, the two pump bodies are cast in one. Two partitions on the center lines at right angles (see Fig. 57) divide this body into four chambers. In the larger sizes, 12 inches and over, there are two separate pump bodies arranged side by side and connected by pipes, as in Fig. 58.

The suction valves in this type of pump are located in the valve deck underneath the plunger, and the discharge valves in a valve deck above the plunger. In order to do the necessary machine-work on the valve decks, it is necessary to part the pump body somewhere, and the best place to do so is just above the discharge valve deck. Then the holes in this upper deck can easily be reamed out and tapped while those in the lower deck can be reached by means of a long shank. On very large pump bodies, it is desirable to divide the body into more than two parts, and then another joint is made right over the suction valve. This introduces a feature which is not very desirable, since an important part of a pump should never rest on a gasket. Any joint, even though carefully made, is liable to leak sometimes, and this would mean the dismantling of the entire pump in order to renew the gasket.

The plunger is made of cast iron, as a rule, and works in a bronze ring with a smooth bore. Water grooves, that are supposed to prevent leakage by changing the velocity of the water passing through, fail to fulfill this purpose, because the water has not time to change its velocity in this short distance; besides, these grooves form receptacles for dirt and sand and are therefore more of a detriment than a help. Good workmanship demands that the clearance between the plunger and the ring should not exceed 0.001 inch all around.

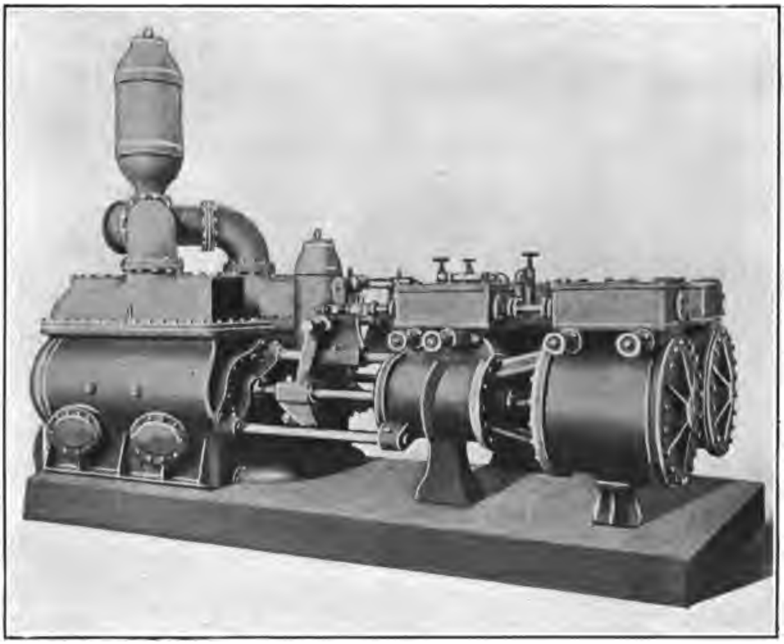


FIG. 58.—Straightway pump with separate bodies.

For general service these straightway pumps are built for pressures up to 150 pounds per square inch.

55. Low-service Straightway Pumps.—The same type is used for tank pumps, circulating pumps and all services where the pressure does not exceed 75 pounds per square inch. For this light pressure the two pump bodies may be cast in one piece up to 16-inch diameter of plunger.

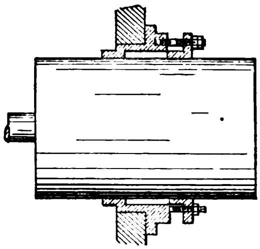


FIG. 59.—Inside-packed plunger.

Instead of using a piston and sleeve some engineers prefer an inside-packed plunger. Here the plunger ring is enlarged so as to form a stuffing-box which is fitted with a gland, as in Fig. 59. As the gland is subjected to external pressure it must be locked by two or three setscrews, otherwise the water pressure will compress the packing to such an extent as to do damage.

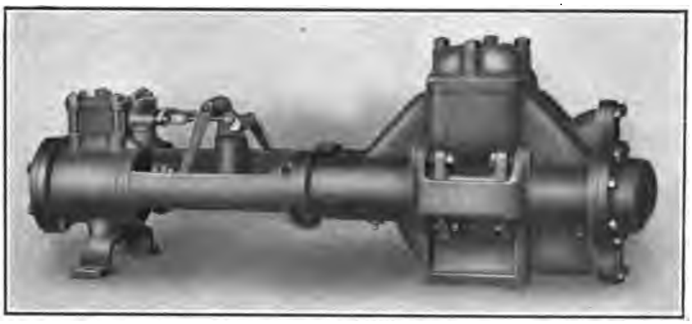


FIG. 60.—Small submerged outside center-packed pump, valve-plate pattern.

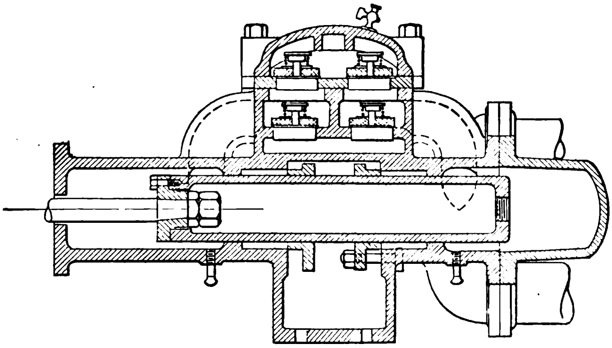


FIG. 61.—Small submerged outside center-packed plunger pump.

56. Outside Center-packed Pumps.—For pressures higher than 150 pounds and up to 300 pounds the pumps illustrated in Figs. 60, 61, and 62 are recommended. The two former are of the submerged plunger type and the latter is built on the same lines as the straightway pattern, except that in the center between the two pump chambers there is an open space and each chamber is provided with a stuffing-box through which the plunger works.

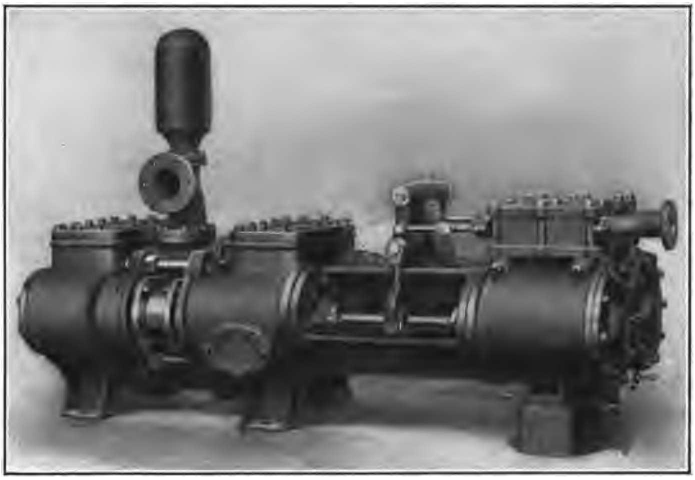


Fig. 62.—Outside center-packed plunger pump.

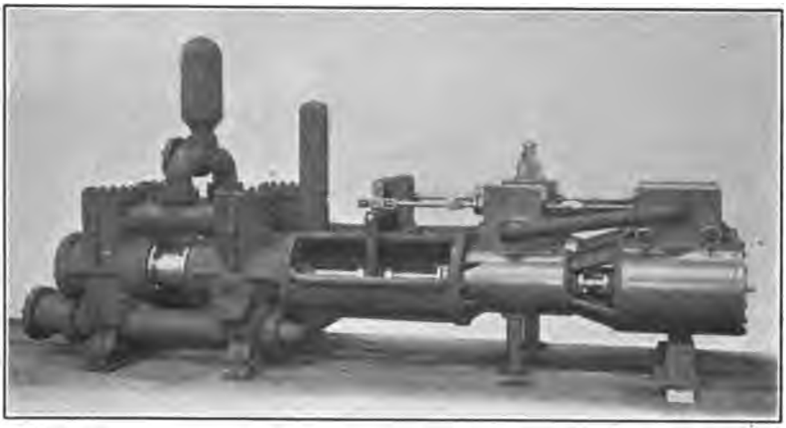


Fig. 63.—Outside center-packed plunger pump with separate pump bodies.

These pumps, while considered the best, are also the most costly. The plunger leakage can be detected and stopped while the pump is in operation, and the plungers can be watched and any cutting readily seen from the outside. Frequently a grain of sand is caught by the packing and, if not removed, will cut a deep groove into the plunger. The plungers should be of hard cast iron and polished in order to preserve the packing.

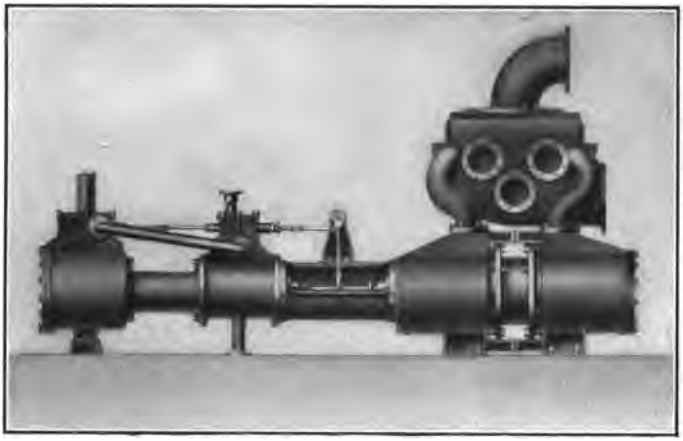


FIG. 64.—Large submerged, outside center-packed pump.

In this type of pump, there are four single-acting pump chambers which, for convenience in manufacture, may be combined in various ways. First, they may be in a single casting,

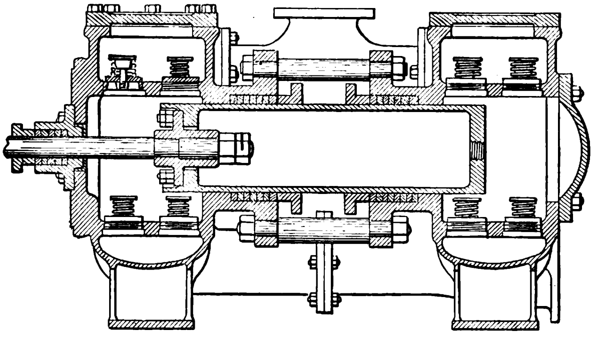


FIG. 65.—Outside center-packed pump, twin pattern and rod attached to end of plunger.

as in Figs. 60 and 61. On larger sizes, however, it is preferable to sectionalize them, in which case they may be split across the axis of the pump, as in Fig. 62. Here the two inboard pump bodies are in a single casting and, likewise, the two outboard pump bodies. Again, they may be split in the axis of the pump, in which case there are two separate double-acting pump bodies each in a single casting arranged side by side and connected by pipes. This type is illustrated by Fig. 63. On large pumps, each single-acting pump body may form a separate casting.

A low-service, outside, center-packed pump is shown by Fig. 64. Here the valve chamber of the submerged piston pump is used and mounted on packed pump barrels.

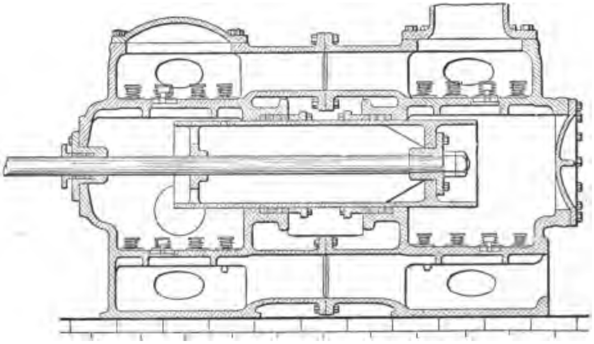


Fig. 66.—Outside center-packed pump, four separate pump bodies and rod extending through plunger.

Figs. 65 and 66 are sectional elevations through the pump body, showing two methods of fastening the plunger rod, which are self explanatory.

57. Pressure Pumps.—With pumps intended for pressures over 300 pounds, extra precautions must be taken. All the castings must be reduced to the smallest possible dimensions and, wherever feasible, must be of cylindrical form, which offers the greatest strength with the least material. Therefore, the pump bodies are cast as plain round barrels, provided with the necessary openings on top for attaching the valve chambers (see Fig. 67).

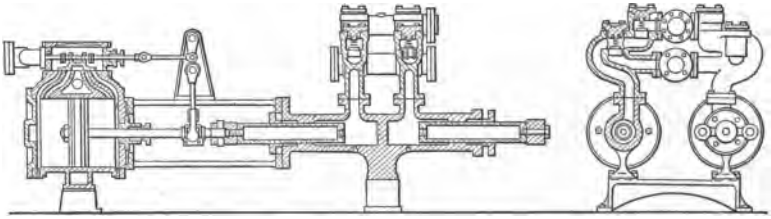


Fig. 67.—Sections of pressure pump.

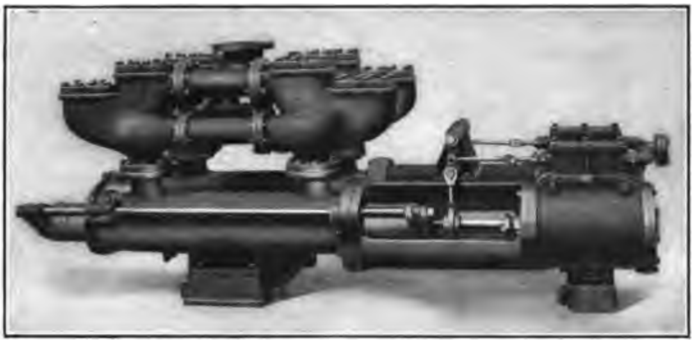


FIG. 68.—Outside end-packed pressure pump.

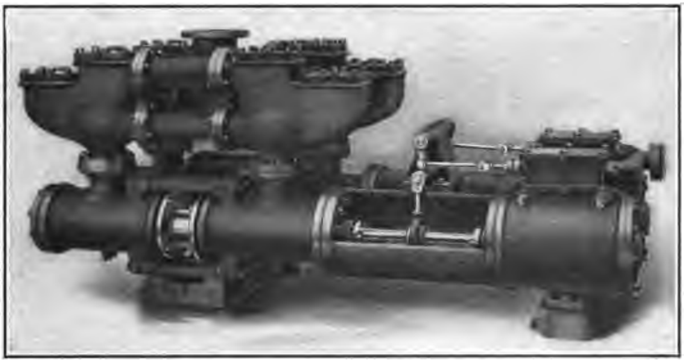


FIG. 69.—Outside center-packed pressure pump.

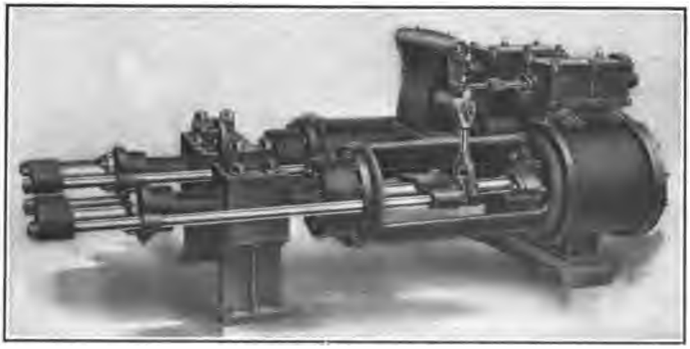


FIG. 70.—Pressure pump with forged pump end.

The valve chambers are small and are equipped with either one pair of valves each, or with two pairs as in Figs. 68 and 69, where the double cover indicates two valves. These pumps may be made with end-packed plungers, as in Fig. 68, or with outside center-packed plungers as in Fig. 69.

The former type has four single-acting plungers and requires two crossheads on each side. These are connected by steel rods called *side rods.*

The latter type has a center rod attached to the plunger, there being only one double-acting plunger on each side. Cast iron is used in building such pumps for pressures up to 1000 pounds per square inch and cast steel up to 3000 pounds per square inch. For pressures higher than these, the pump should be made of forged steel. Such a pump is shown in Fig. 70. Here the valve chamber is a solid block of steel, and the valve ports and the necessary cavities for receiving the valves are drilled out. The barrels may be forged in one piece with this block, or they may be steel castings, in which case they are provided with flanges for bolting to the valve chamber; see Fig. 71. Cast steel is entirely satisfactory for such simple castings.

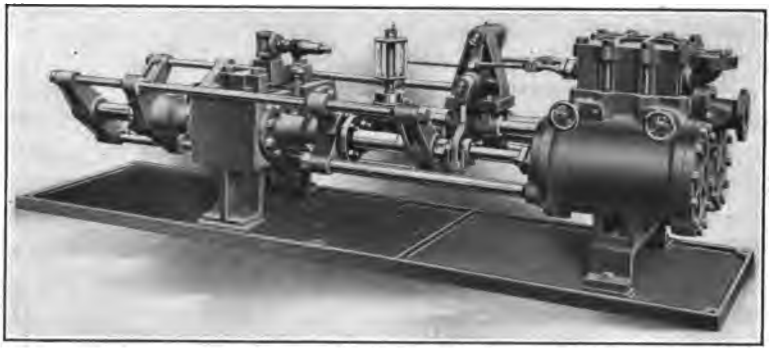


FIG. 71.—Pressure pump with forged valve chamber.

58. Ram Pattern Pumps.—For prospecting and similar purposes, it was found desirable to use pumps of small size which combined the advantages of a pressure pump with the low cost of a submerged piston pump. For this purpose a new type, called the *ram pattern pump,* was developed. This is illustrated in section in Fig. 72. It has four single-acting plungers connected by side rods, just like a regular end-packed pressure pump, but instead of using expensive valve chambers, the center part is designed similar to the valve box of a submerged piston pump with a loose valve plate and cap.

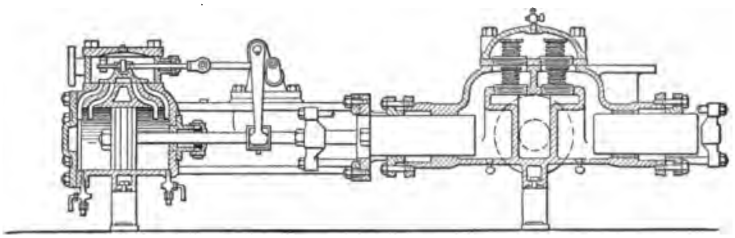


FIG. 72.—Ram pattern pump.

59. Simple Cylinder Pumps.—This type of direct-acting pump has only one, the high pressure, steam cylinder to each water cylinder, and for this reason it is generally called a *high-pressure pump.* As this term is frequently misunderstood to mean a pump that works against a high water pressure, in the present discussion pumps of this type will be designated as *simple-cylinder pumps,* to distinguish from the multi-stage pumps having more than one steam cylinder to each water cylinder.

As the steam acts with full pressure during the entire stroke, there is neither cutoff nor compression, nor other features met with in a steam engine diagram. The indicator diagram then becomes a plain rectangle, as shown in Fig. 73, the steam line being a straight line at the absolute pressure p and the exhaust line a straight line at the pressure b. Boiler pressure + 10 pounds = p, which allows 4.7 pounds for wire-drawing. The back pressure b is usually taken at 16 pounds absolute.

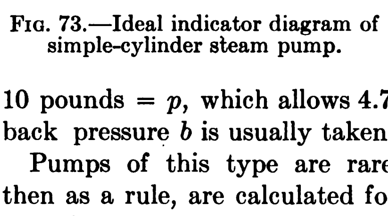


FIG. 73.—Ideal indicator diagram of simple-cylinder steam pump.

Pumps of this type are rarely used condensing and, even then as a rule, are calculated for running noncondensing. Frequently they exhaust against a back pressure, as in cases where the exhaust steam is utilized for industrial purposes. The mean effective pressure is $p - b$.

60. Compound Pumps.—New problems loomed up when it became necessary to devise some means for attaining greater economy in the direct-acting pump. Pumps with simple steam cylinders, such as previously considered, cannot utilize the expansive force of the steam and there is no flywheel to store up the energy during the first half of the stroke in order to replenish it during the remainder of the stroke.

If a steam diagram is drawn of an ordinary cutoff engine, Fig. 75, it will be seen that the steam force decreases from the point of the cutoff B to the end of the stroke C. The steam line consists of two parts, namely, the period of admission, represented by the line AB, and the period of expansion, represented by a curve BC. Considering that the load in a pump is constant, it may be represented by a straight line FH, which coincides with the mean effective pressure line of the steam diagram. This line then represents the total load of the pump, namely, the plunger load plus friction.

Thus in the cutoff diagram there is an excess of energy during the first half of the stroke, represented by the area of $FABG$, and a deficiency during the latter half of the stroke, represented by the area GHC, and it is evident that such a machine will not run without some compensating device, such as a flywheel. The piston of a direct-acting pump without any such device will stop the moment the steam is cut off, except, perhaps, in fast-running pumps with heavy reciprocating parts, where it may be possible to cut off a short distance from the end of the stroke, the momentum of the reciprocating parts carrying the piston to the end. In ordinary pumps for general service, however, this is not practical and steam must be admitted the full length of the stroke, and exhausted into the atmosphere or condenser at high pressure, which means a great waste of fuel.

However, at least part of this waste may be saved by the addition of another and larger steam cylinder into which the steam can expand after having done part of its work in the smaller cylinder; the machine is then called a *compound pump*, such as illustrated in Fig. 74. This shows two steam cylinders—a small one, the high-pressure cylinder, and a larger one, the low-pressure cylinder—arranged in tandem, side by side and each pair attached to a pump.

In the early days of the direct-acting pump an attempt was made to get along with two steam cylinders, a high-pressure on one side and a low-pressure on the other. This, however, was not successful on account of the differences between the steam forces in the two sides. Although it was possible to equalize them for one set of conditions they were not equal for any other set, and it is one of the requirements of a reliable pump that it must operate under a wide range of conditions such as speed, pressures, vacuum, etc. This type, called the *cross-compound pump*, was therefore soon abandoned (see Fig. 31).

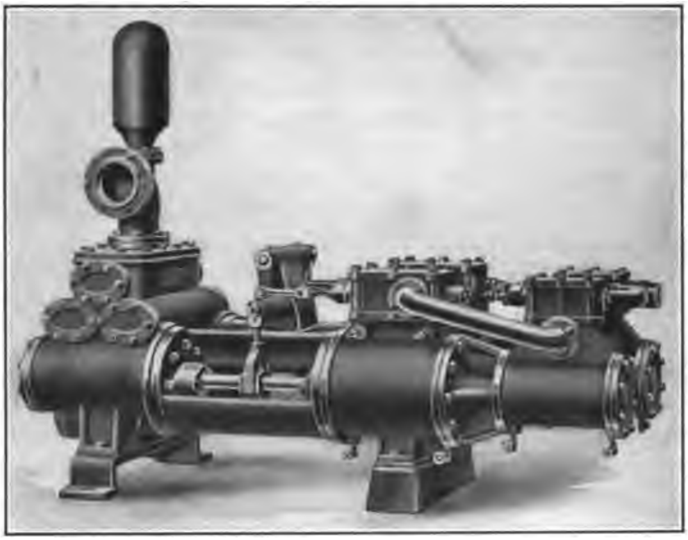


Fig. 74.—Compound pump with overhanging high-pressure cylinders.

In a compound pump, for the reasons before stated, steam must be admitted for the full length of the stroke in the high-pressure cylinder. Therefore, this part of the steam diagram is characterized by a straight line *AB* (Fig. 76). At point *B* the exhaust valve opens, and the steam blows through the pipes and the low-pressure steam chest into the clearance space of the low-pressure cylinder.

In Figs. 77 and 78 the two significant positions of the pistons are shown, the *LH* side being indicated in full lines with the *RH* side added in dotted lines. In exhausting, the pressure of the steam is reduced somewhat; it has expanded into a larger volume but without having done any useful work. Assume that it has expanded to the pressure *C*, Fig. 76, which acts as a back pressure against the high-pressure piston and urges on the low-pressure piston. The difference between the two forces causes the pistons to move and by this motion the space occupied by the steam is increased, since the volume in the high-pressure cylinder diminishes, while that in the low-pressure cylinder increases; the latter being the larger, increases at a faster rate. Therefore, at the end of the stroke, the volume is larger than at the beginning, and the steam has expanded from one cylinder into the other.

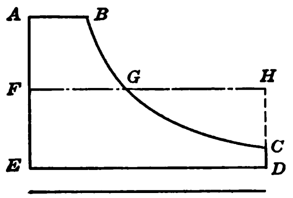


FIG. 75.—Ideal indicator diagram of cut-off engine.

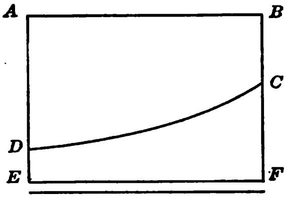


FIG. 76.—Ideal indicator diagram of compound direct-acting steam end.

In expanding, however, it has not followed the regular expansion curve, as shown in Fig. 75, but instead a much flatter curve, having its zero point some distance away from that of the regular diagram. This zero point is easily located without having to resort to a complicated formula, as will be explained in the following example:

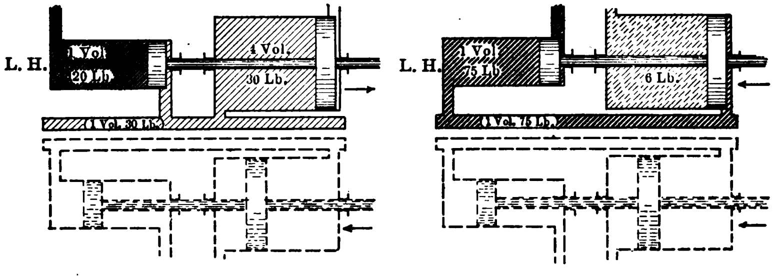


FIG. 77. FIG. 78.

Position diagram of compound pump.

Assume a 12 and 24 × 8½ × 18-inch compound duplex pump, which means that the steam end consists of two high-pressure cylinders 12 inches in diameter and two low-pressure cylinders 24 inches in diameter, one high- and one low-pressure cylinder being arranged tandem on each side; also that this steam end is attached to a duplex pump end with 8½-inch plungers, and that the common stroke of all is 18 inches. Let the boiler pressure be 110 pounds gage, and the low-pressure cylinder exhaust into a condenser at a vacuum of 26 inches of mercury.

The first thing is to find the ratio of the two cylinders. Neglecting the areas taken up by the piston rods, the ratio is

$$R = \frac{24^2}{12^2} = 4$$

Next, find the volume of the intermediate space between the two cylinders of one side, consisting of the high-pressure ports, the connecting pipes, the low-pressure steam chest, and the low-pressure ports. Assume that this has been calculated and found to equal one high-pressure cylinder volume.

The steam in passing from the boiler through the steam pipe, the throttle valves and the steam ports will lose pressure by friction, which loss is commonly called *wire-drawing*. In laying out indicator diagrams absolute pressures must always be dealt with; therefore, a wire-drawing of 4.7 pounds will be assumed, which in general would not be far from the truth. With the boiler pressure 110 pounds gage or 124.7 absolute, this would make the absolute initial pressure in the high-pressure cylinder 120 pounds. In other words, 10 pounds is added to the boiler pressure to obtain the absolute initial pressure in the high-pressure cylinder.

When the steam valve opens and steam is admitted into the high-pressure cylinder, it first fills the clearance space, and its condition is indicated by the point *A*, Fig. 76. The piston now travels to the right, with steam at full pressure behind it, and this constant pressure will show as a straight line until the piston has reached the end of the stroke or point *B*. The line *AB* then represents the steam at 120 pounds acting on the high-pressure piston during admission. At *B* the exhaust valve opens and the steam exhausts into the intermediate space where it encounters and mixes with a certain amount of steam left over from the preceding stroke; this condition is indicated by Fig. 78. As is shown in Fig. 77, the intermediate space, just before the valve shifted, was in communication with the low-pressure cylinder which is ready to exhaust into the condenser. It is clear that more steam cannot be exhausted than is taken in; therefore, the weight of steam in the low-pressure cylinder at the end of the stroke must be the same as that in the high-pressure cylinder at the end of the stroke.

The calculations to follow will be based on the assumption that the steam, in passing through the engine, follows the law

$$pv = Constant$$

This law, although not theoretically correct, has been found to be a much closer approximation to what actually occurs in the cylinder than the saturation curve, and it may be well to make a comparison between these two curves.

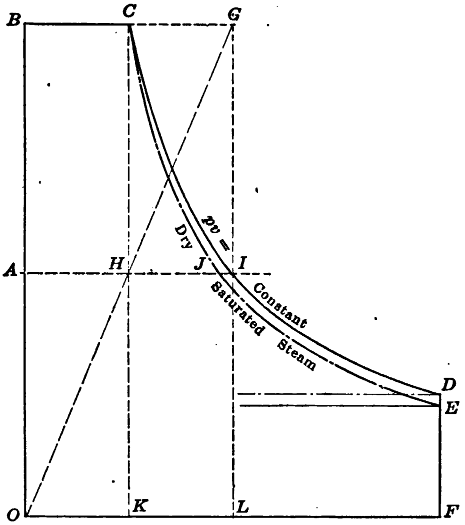


FIG. 79.—Comparison of *pv* constant curve and dry saturated steam curve.

If the curves representing the two conditions are plotted, they will be found to lie close together; see Fig. 79. The upper one is the *pv* curve, which is an equilateral hyperbola. When used in connection with the compression of permanent gases, this curve is called the *isothermal,* or curve of constant temperature, which name does not, of course, apply when used for steam. It is constructed readily by the following method:

With *OBCK* given, draw a vertical such as *LG*; then diagonal *OG* and a horizontal line through the intersection point *H*. Where this line intersects with line *LG* will be a point *I* of the *pv* curve.

The saturation curve can be plotted directly from the steam tables (Marks & Davis) by the following method:

Let the absolute initial steam pressure be 120 pounds and base the calculations on 1 pound of steam. From the tables the specific volume of 1 pound of steam at 120 pounds pressure is 3.726, which means that, at point *C*, the steam occupies 3.726 cubic feet. Now let the piston travel forward until the volume has expanded to double the original, or 7.452 cubic feet. Then according to the law, $pv =$ constant, the pressure has dropped to 60 pounds. Consulting the steam tables, it will be found that at a pressure of 60 pounds the specific volume is only 7.17 cubic feet per pound, or less space than is indicated by point *I*, which represents 7.452 cubic feet. Hence, a new point *J* is located somewhat inside the *pv* curve. In this manner, it is easy to plot the entire saturation curve.

The steam, although it may be dry when entering the cylinder, comes in contact with the cold walls which have been in contact with the exhaust steam, cools off and naturally precipitates some moisture. Therefore, it is evident that the steam at the point of cutoff is not dry, but will contain a certain percentage of water at high temperature which raises its density. As the steam expands, it becomes colder and immediately the hot water begins to evaporate and occupy more space, with the result that the actual curve is raised up to, and often above, the *pv constant* curve. We are therefore justified in using this simple and convenient law as the basis for calculations.

To simplify matters, all volumes will be referred to that of the high-pressure cylinder which will be designated as "one volume." Then at the end of the high-pressure stroke, the volume $(v) = 1$, the pressure $(p) = 120$ and the product $(pv) = 120$ which is constant.

This figure must also hold true for the steam exhausted by the low-pressure cylinder, but here the volume is four times as large, or $v = 4$; whence, in order to satisfy the equation, $pv = 120$, p must equal 30. That is, the steam at the terminal point *D*, Fig. 76, is 30 pounds and since the intermediate space is in communication with this steam, the pressure therein must also be 30 pounds as represented.

Therefore, the high-pressure steam in exhausting meets one volume of steam at 30 pounds and we have then a total amount as follows:

High-pressure cylinder (one volume)	1 × 120 =	120
Intermediate space (one volume)	1 × 30 =	30
Total (two volumes)		150

Hence the pressure of the mixture (see Fig. 78) must be 150 ÷ 2 = 75 pounds. This is also indicated by point *C* in the diagram, Fig. 76.

Two points *C* and *D* of the intermediate expansion curve have now been established, and by the same method, a number of other points could be located. This, however, can be accomplished graphically in a simpler manner. Having two points, *C* and *D*, we draw horizontal lines through them as shown in Fig. 80 and connect the two opposite points, *G* and *H*, by a straight line. Find the zero point *O* at the intersection of this diagonal and the line of absolute zero. Having located the zero point, a number of points such as *K* can be found readily by the method of similar triangles, as already explained. Adding a back pressure line, at 6 pounds for instance, completes the diagram.

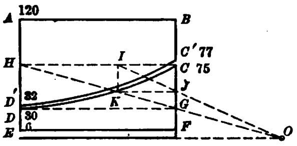


Fig. 80.—Method of plotting expansion curve.

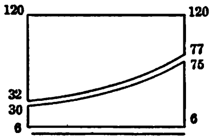


Fig. 81—Ideal indicator diagram with cross exhaust closed.

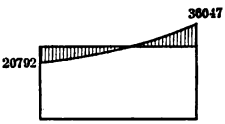


Fig. 82.—Propulsion diagram of compound pump.

On its passage from the high-pressure cylinder through the intermediate space to the low-pressure cylinder, the steam had to pass many tortuous ports, and has suffered loss by friction, which can be assumed as amounting to 2 pounds pressure; this will be represented by another line *C′D′*, parallel to *CD* and 2 pounds above. The upper line *C′D′* is now the back-pressure line for the high-pressure cylinder and the lower one, *CD*, is the low-pressure admission line. The theoretical diagram is now complete and it differs from the actual diagram in that it does not show the round corners and other imperfections. Fig. 81 shows the diagram as computed.

Analyzing this diagram, we have:

High-pressure	Beginning of stroke	120 − 77 = 43 lb.
	End of stroke	120 − 32 = 88 lb.

$$\text{Low-pressure}\left\{\begin{array}{lr}\text{Beginning of stroke} & 75 - 6 = 69 \text{ lb.}\\ \text{End of stroke} & 30 - 6 = 24 \text{ lb.}\end{array}\right.$$

The areas of the two cylinders are:

Area of high-pressure cylinder = 113 sq. in.
Area of low-pressure cylinder = 452 sq. in.

and the actual steam forces will be:

$$\text{High-pressure}\left\{\begin{array}{rl}113 \times 43 = 4{,}859 & \text{Beginning}\\ 113 \times 88 = 9{,}944 & \text{End}\end{array}\right.$$

$$\text{Low-pressure}\left\{\begin{array}{rl}452 \times 69 = 31{,}188 & \text{Beginning}\\ 452 \times 24 = 10{,}848 & \text{End}\end{array}\right.$$

Therefore, at the beginning of the stroke there is a total force of

$$4859 + 31{,}188 = 36{,}047 \text{ lb.}$$

and at the end

$$9944 + 10{,}848 = 20{,}792 \text{ lb.}$$

or 15,255 pounds less at the end than at the beginning of the stroke. This condition is represented in Fig. 82, where the shaded areas represent the unbalanced forces. It is quite a drop, and almost as bad as if there had been a cutoff in the cylinder.

It is safe to predict that this pump would not run, except, perhaps, if the reciprocating parts were very heavy and the speed high.

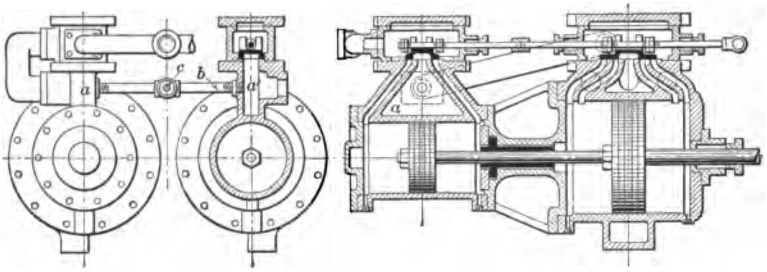


FIG. 83.—Sections of compound steam end showing cross exhaust and floating sleeve.

61. Cross Exhaust.—The difficulty just mentioned was overcome by installing the cross exhaust which is a connection between the two high-pressure exhaust pipes, with a valve inserted for regulating purposes, called the *cross-exhaust valve.* This is seen in Fig. 83, in which *a* and *a'* are the high-pressure exhaust passages connected by a pipe *b* with the valve *c* in the center.

Assume that this valve is full open, and that the communicating pipe is so large as to always have the same pressure in the two passages *a* and *a'*; then go through the same cycle as before and note the following:

The line of admission is the same as before, namely, a straight line *AB* at 120 pounds, Fig. 84. At the end of the stroke, corresponding to point *B*, there is in the high-pressure cylinder a quantity of steam that can be expressed by the product of volume and pressure, or

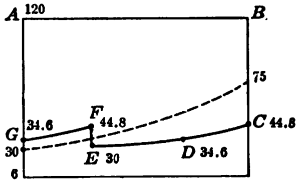


FIG. 84.—Ideal indicator diagram showing effect of cross exhaust.

$$pv = 120 \times 1 = 120$$

This is indicated in Fig. 86 by the left-hand cylinder shaded in heavy lines. Fig. 85 shows the beginning of the stroke of the left-hand side and Fig. 86 the end of the stroke. Assume that when the left-hand piston has reached the end of its stroke, the right-hand piston has arrived at the position shown in Fig. 86. The valves on the left-hand side now reverse and the charge of 120 flows into the intermediate space which has been, a moment before, in communication with the low-pressure cylinder, at the point of exhausting, when there was in the cylinder a pressure of 30 pounds.

The space into which the left-hand, high-pressure cylinder exhausts consists of the following parts:

Left-hand intermediate space	1 vol.
Right-hand intermediate space	1 vol.
Right-hand high-pressure cylinder	0.3 vol.
Right-hand low-pressure cylinder	2.8 vol.
Total	5.1 vol.

of steam at 30 pounds. The amount of steam contained in this space can be expressed by 5.1 × 30 = 153.

This steam now mixes with the 120 units coming from the high-pressure cylinder, making a total of 120 + 153 = 273, which is the constant for this engine; that is, $pv = 273$, which represents the steam that is always present in the engine, and 120 represents the charge or the steam that is passing through the engine doing the useful work.

It is now easy to determine the pressure at any point by merely calculating the volume and dividing into 273. In Fig. 87 this total volume is:

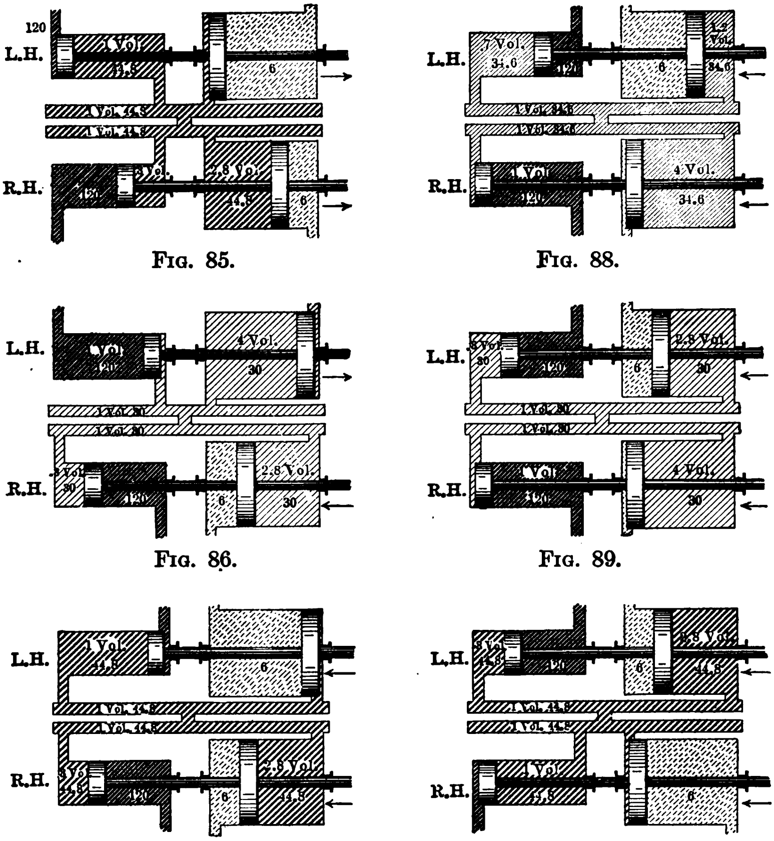


Fig. 85. Fig. 88.

Fig. 86. Fig. 89.

Fig. 87. Fig. 90.

Figs. 85–90.—Position diagrams of pistons of compound pump with cross exhaust open.

Left-hand, high-pressure cylinder	1 vol.
Total spaces as above figured	5.1 vol.
Total	6.1 vol.

$$\frac{273}{6.1} = 44.8 \text{ lb., the pressure at } C\text{, Fig. 84.}$$

The left-hand pistons now start and travel three-tenths of the stroke to the position shown in Fig. 88, while the right-hand pistons travel an equal distance and complete their stroke. We have, then, the following spaces:

Left-hand, high-pressure cylinder	0.7 vol.
Left-hand, low-pressure cylinder	1.2 vol.
Left-hand intermediate space	1 vol.
Right-hand intermediate space	1 vol.
Right-hand, high-pressure cylinder	0 vol.
Right-hand, low-pressure cylinder	4 vol.
Total	7.9 vol.

Hence the pressure at point *D*, Fig. 84, is:

$$\frac{273}{7.9} = 34.6 \text{ lb.}$$

or, the steam has expanded from 44.8 to 34.6 pounds following a certain expansion curve which can easily be constructed from *C* to *D*.

The next point of interest is the point *E*, corresponding to the positions of Fig. 89. Here the steam occupies the following spaces:

Left-hand, high-pressure cylinder	0.3 vol.
Left-hand, low-pressure cylinder	2.8 vol.
Left-hand intermediate space	1 vol.
Right-hand intermediate space	1 vol.
Right-hand, high-pressure cylinder	0 vol.
Right-hand, low-pressure cylinder	4 vol.
Total	9.1 vol.

and the pressure at point *E* will be:

$$\frac{273}{9.1} = 30 \text{ lb.}$$

At this instant, the valve is shifted and the cylinder ends are connected, as indicated in Fig. 90. The effect is that 120 units of steam are pushed into the intermediate spaces by the right-hand, high-pressure cylinder, and 120 units of steam are taken out by the right-hand, low-pressure cylinder. But the former had the condition of, $1 \times 120 = 120$, while $4 \times 30 = 120$ represents the latter. This effects a change, as can be proven by calculating the total volume. Referring to Fig. 90:

Left-hand, high-pressure cylinder	0.3 vol.
Left-hand, low-pressure cylinder	2.8 vol.
Left-hand intermediate space	1 vol.
Right-hand intermediate space	1 vol.
Right-hand, high-pressure cylinder	1 vol.
Right-hand, low-pressure cylinder	0 vol.
Total	6.1 vol.

Hence the pressure at point *F*, Fig. 84, is $\frac{273}{6.1} = 44.8$ lb.

It will be noted that the pressure suddenly rises from 30 to 44.8, which is the same as that found for point *C*. This is not surprising when it is considered that the right-hand side in Fig. 90 corresponds with the left-hand side in Fig. 87.

Likewise, the point *G*, Fig. 84, corresponds with point *D*, the pressure being 34.6 pounds when the pistons have reached the end of the stroke. Here the left-hand pistons pause while the right-hand pistons travel on, extending the space and expanding the steam down to 30 pounds, when live steam is admitted, and the pressure is brought up again to 120 lb. as represented by point *A*.

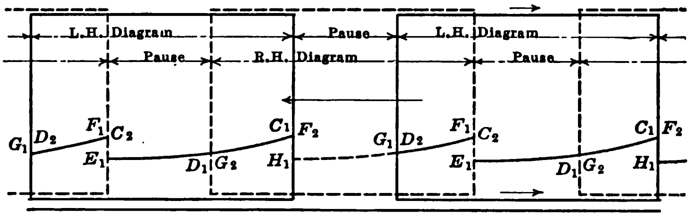


Fig. 91.—Continuous diagram showing effect of cross exhaust.

The analogy of the two sides can be represented nicely by a continuous diagram, as shown in Fig. 91. The letters with subscript "1" refer to the left-hand side, and those with subscript "2," to the right-hand side. The diagram for the *L. H.* side is shown in full, and that for the *R. H.* side in dotted lines. The intermediate line shows clearly the variation of pressures in the intermediate spaces.

The peculiar shape of an indicator diagram of a duplex direct-acting pump, with the cross exhaust full open, can now be observed, and the actual steam forces at each point will be calculated. To do this, obtain from Fig. 84 the forward pressure and deduct therefrom the back pressure. In doing so care must be taken to select two corresponding points, namely, two points equidistant from the beginning of the stroke. *A* represents the beginning of the stroke and *B* the end; then in exhausting, *C* is the beginning and *G* the end. Therefore, the back pressure at *C* must be deducted from the forward pressure at *A*, and likewise, the back pressure at *G* from the forward pressure at *B*.

In the simple case under consideration, error is unlikely since *A* equals *B*, but it is well to understand this point, for in dealing with actual diagrams *AB* may not be a line of uniform pressure.

Again, allowing a loss of 2 pounds between the two cylinders, we find the following forces:

High-pressure Cylinder

Point *C*, 120 − 46.8 = 73.2, on 113 sq. in. = 8272 lb.
Point *D*, 120 − 36.6 = 83.4, on 113 sq. in. = 9424 lb.
Point *E*, 120 − 32.0 = 88.0, on 113 sq. in. = 9940 lb.
Point *F*, 120 − 46.8 = 73.2, on 113 sq. in. = 8272 lb.
Point *G*, 120 − 36.6 = 83.4, on 113 sq. in. = 9424 lb.
Point *H*, 120 − 32.0 = 88.0, on 113 sq. in. = 9940 lb.

Low-pressure Cylinder

Point *C*, 44.8 − 6 = 38.8, on 452 sq. in. = 17,538 lb.
Point *D*, 34.6 − 6 = 28.6, on 452 sq. in. = 12,927 lb.
Point *E*, 30.0 − 6 = 24.0, on 452 sq. in. = 10,848 lb.
Point *F*, 44.8 − 6 = 38.8, on 452 sq. in. = 17,538 lb.
Point *G*, 34.6 − 6 = 28.6, on 452 sq. in. = 12,927 lb.
Point *H*, 30.0 − 6 = 24.0, on 452 sq. in. = 10,848 lb.

Adding the corresponding forces,

Points *C* and *F*, 8272 + 17,538 = 25,810 lb.
D and *G*, 9424 + 12,927 = 22,351 lb.
E and *H*, 9940 + 10,848 = 20,788 lb.

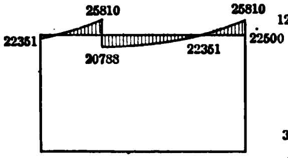


Fig. 92.—Propulsion diagram with cross exhaust open.

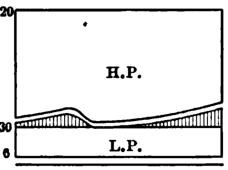


Fig. 93.—Indicator diagram with cross exhaust open.

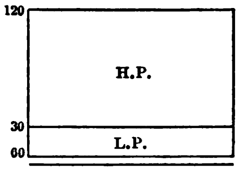


Fig. 94.—Simplified diagram with cross exhaust open.

These forces laid off graphically will give a nearly uniform line, as shown in Fig. 92. It is now easy to draw a line of average force which is found at 22,500 pounds and represents the average theoretical steam force exerted by the steam end.

In the actual diagrams, there will be no such sudden changes of pressures, as have been assumed in the theoretical examples. It takes time for the steam to blow from one space to the other, and the corners will be more or less rounded, as in Fig. 93. The irregularities that remain are easily overcome by the inertia of the reciprocating parts. By this rounding out some more unavoidable losses are introduced, for which an allowance of about 7 per cent. may be made. Therefore, the actual steam force would be

22,500 − 7 per cent. = 21,000 lb. (approximately)

Going a step further to simplify the diagram, draw a straight line at 30 pounds (Fig. 93); assume this to be the intermediate line. The shaded area has now been taken away from the large cylinder and added to the small cylinder. The diagram, then, does not represent the entire energy and, to make up for this error, neglect the losses between cylinders, which will now give an extremely simple diagram consisting of two plain rectangles, as shown in Fig. 94.

Using this as a basis, we have:

High-pressure cylinder	(120 − 30)	= 90 on 113 sq. in.	= 10,170 lb.
Low-pressure cylinder	(30 − 6)	= 24 on 452 sq. in.	= 10,848 lb.
Total			21,018 lb.

or practically the same as before.

Therefore, it is unnecessary to go through such a laborious calculation as has been described, since one can arrive at the same result by simple means. Referring to Fig. 95, let

p = Absolute initial pressure;
R = Ratio of cylinders;
b = Back pressure;
A = High-pressure cylinder area.

then, referring all mean effective pressures (m.e.p.) to the area of the high-pressure piston,

$p - \frac{p}{R}$ = m.e.p. on high-pressure piston.

$\frac{p}{R} - b$ = m.e.p. on low-pressure piston.

$R\left(\frac{p}{R} - b\right) = p - Rb$ = m.e.p. on low-pressure piston referred to the area of the high-pressure piston. These two added together give the total mean effective pressure referred to the area of the high-pressure piston; that is,

$$\text{m.e.p.} = p - \frac{p}{R} + p - Rb = 2p - \frac{p}{R} - bR$$

It is customary to make the back pressure b equal 6 pounds for condensing engines and 16 pounds for noncondensing engines.

A comparison of the diagram thus found for a compound pump with that of a simple-cylinder pump will show the gain in economy. The size of the compound pump, it will be remembered, was 12 & 24 × 8½ × 18 inches, and the total steam force was found to be 21,000 pounds. A simple pump, under the same conditions, would require a large cylinder and the m.e.p. would be 120 − 6 = 114 pounds. Dividing this into the total steam force of 21,000 pounds would give 184 square inches as the area of the piston. This corresponds to a 15¼-inch cylinder, while in the compound pump at every stroke a 12-inch cylinderful of steam at the same pressure (120 pounds) is used, which would be equivalent to a saving of nearly 40 per cent. with the compound pump. At normal speeds the saving will tend to be more, for then the cross-exhaust valve can be partly closed; however, other conditions will tend to reduce this gain. In practice, a saving of the compound over the simple pump of from 25–30 per cent. has been demonstrated by actual tests.

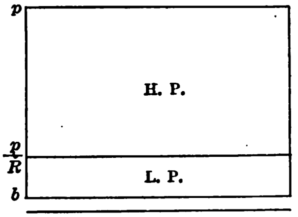


FIG. 95.—Simplified diagram of compound steam end.

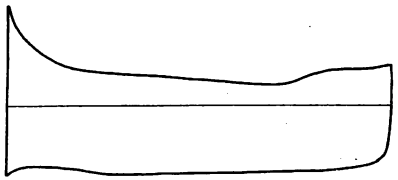


FIG. 96.—Actual indicator diagram with cross exhaust partly open.

An indicator diagram taken off the low-pressure cylinder of a compound pump exhibiting the peculiar feature of a hump near the end of the stroke is shown in Fig. 96.

The cross-exhaust valve should never be opened more than is necessary to insure a full stroke, because opening of this valve is always accompanied by loss in power. In Fig. 84 a dotted line, sloping from 75 to 30 pounds, indicates the condition with the cross-exhaust valve closed, and the irregular line shown in full indicates the condition with the cross-exhaust valve open. It will be noted that by opening the valve some steam is taken away from the large cylinder and added to the small cylinder, thereby reducing the resulting steam forces. The difference is also shown by the computations, there being an average steam force of 25,000 pounds with the cross exhaust closed, while with it open the average force was only 21,000 pounds—a loss of 4000 pounds due to the opening of the valve.

This loss cannot be avoided. With the cross-exhaust valve closed the pump will short-stroke and the losses caused thereby would exceed those caused by the opening of the cross exhaust considerably. If the pump is running properly and the cross exhaust is then closed it will immediately speed up, but at the same time its stroke will shorten.

62. Ratio of Cylinders.—A simple diagram was developed in Fig. 95, which represents fairly well the distribution of energy in a compound direct-acting pump. In order to draw this, however, one must know the ratio (*R*) of the two cylinders. To do this properly the two indicator diagrams must be combined into one by using a base of volumes instead of the stroke. The low-pressure diagram will then be *R* times as long as the high-pressure diagram and a combined diagram will be obtained as is illustrated in Fig. 97*a*. The points *C* and *D* are the only ones that belong to the curve of expansion, which is indicated by the broken line. The line *ABCDE* can be considered as representing the diagram of an engine of a stroke *AE*, cutting off at *C*, and the work would then be represented by the area of this figure. In a direct-acting pump, however, only the areas of the two rectangles, *FBCG* and *AFDE*, represent the work done, leaving the triangle *GCD* as the measure of the loss in economy suffered by the absence of a crank and flywheel.

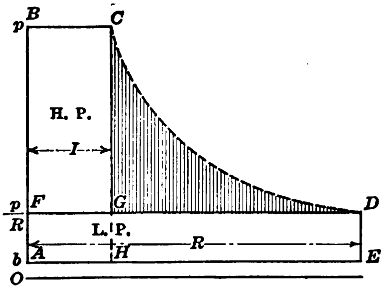


FIG. 97*a*.—Combined diagram of compound steam end.

Let p be the initial pressure, represented by OB, and $\frac{p}{R}$ the back pressure on the high-pressure piston; also the initial pressure in the low-pressure cylinder is OF, and b, the back pressure in the low-pressure cylinder is represented by OA. Then by moving line FD up or down, in other words, by changing the ratio of the cylinders, the area of the triangle, and consequently the losses, can be changed.

To obtain an expression for this loss, draw line HG. The area $HGDE$ now represents the *gain due to compounding*, because, with only one cylinder, the area $ABCH$ would be a measure of the energy. Therefore, it must be the aim to make the rectangle $HGDE$ a maximum by shifting the line FD to a position where this condition exists. Going first to one extreme, let the line FD be moved up until it coincides with BC. Following through the various stages, it will be seen that the area $HGDE$ becomes smaller and smaller and finally zero, under which condition the high-pressure cylinder volume equals that of the low-pressure cylinder and R equals one.

Next, move the line FD down until it coincides with line AE. Here also the rectangle $HGDE$ diminishes until finally it is zero, which condition is represented by $R = \frac{p}{b}$; in other words, the low-pressure volume $= \frac{p}{b}$ times the high-pressure volume. From Fig. 97*a*, the following expression can be obtained:

$$HGDE = (R - 1)\left(\frac{p}{R} - b\right) = p + b - \left(\frac{p}{R} + bR\right)$$

Since p and b are constants, this expression is a maximum when $\left(\frac{p}{R} + bR\right)$ is a minimum.

Designate the expression by y; then knowing that it is a minimum when the differential coefficient of y, regarded as a function of R equals zero,

$$\frac{dy}{dR} = b - \frac{p}{R^2} = 0$$

$$b = \frac{p}{R^2}$$

from which equation is derived the simple and convenient formula:

$$R = \sqrt{\frac{p}{b}}$$

The work done in the high-pressure cylinder which is represented by the area of the rectangle *FBCG*, is

$$\left(p - \frac{p}{R}\right) \times 1 = p - \frac{p}{R}$$

That done in the low-pressure cylinder is represented by the area of the rectangle *AFDE* and equals

$$\left(\frac{p}{R} - b\right) \times R = p - bR$$

Substituting the value, $b = \frac{p}{R^2}$ in this expression, we have area $AFDE = p - \frac{p}{R}$, which is the same as that found for the high-pressure cylinder and means that, under the most economical conditions, the work is distributed equally between the two cylinders.

The formula $R = \sqrt{\frac{p}{b}}$ should be used with caution as it is not merely the ratio of the cylinders that determines the economy and commercial success of an engine; the deciding factor is generally the initial cost, and it is obvious that this expense increases with the size of the low-pressure cylinder. Above a certain point, the gain in economy will be insignificant compared with the greatly increased initial cost, so that it has become the general practice not to exceed the ratio of 4 to 1, corresponding to what is considered the most economical point of cutoff in an engine, namely, ¼. Furthermore, each manufacturer will establish certain standard sizes and will, therefore, be able to offer them at a comparatively low price for all sets

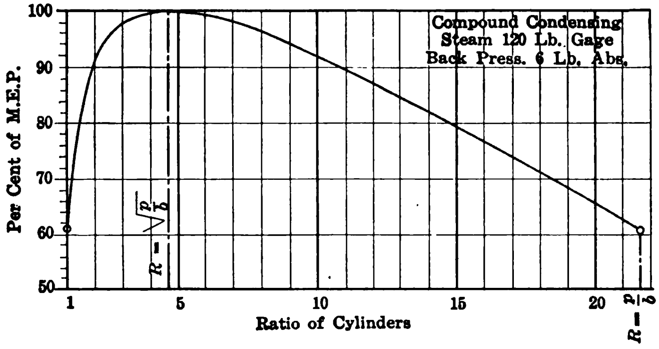


FIG. 97*b*.

of conditions, even if these sizes have ratios below the most economical.

Fig. 97*b* shows the effect on the mean effective pressure and also on the economy, of varying the ratio of cylinders. The diagram shows that a deviation of 25 per cent. in the ratio changes the economy only a few per cent.

TABLE 7.—RATIOS OF CYLINDERS IN A COMPOUND PUMP

$$R = \sqrt{\frac{p}{b}}$$

where $R = \frac{\text{Low-pressure area}}{\text{High-pressure area}}$;

p = Absolute initial pressure;

b = Absolute back pressure = 16 pounds noncondensing and 6 pounds condensing.

Gage pressure in boiler	Absolute initial pressure, p	Noncondensing, R	Condensing, R
60	70	2.09	3.42
70	80	2.24	3.65
80	90	2.37	3.88
90	100	2.5	4
100	110	2.62	4
110	120	2.74	4
120	130	2.85	4
130	140	2.96	4
140	150	3.06	4
150	160	3.16	4
160	170	3.26	4
170	180	3.35	4
180	190	3.44	4
190	200	3.54	4
200	210	3.62	4

DESIGN OF COMPOUND PUMPS

63. Outboard High-pressure Cylinders.—The simplest design is that shown in Figs. 74 and 98. The low-pressure cylinders are located next to the pump end and each high-pressure cylinder is attached to the rear end of its low-pressure cylinder by means of an intermediate head.

The high-pressure cylinder, when small, may be made overhanging and without any support, as shown in Fig. 74, whereas larger high-pressure cylinders, of 15-inch stroke and over, should rest on a foot, as in Fig. 98. The intermediate head may be made with stuffing-boxes, as in Fig. 98; or it may be fitted with a solid floating sleeve, as in Fig. 83. This sleeve is of cast iron, and is held in place by its flange. Both sides of the flange are scraped to an easy steam-tight fit between the adjoining parts and sufficient clearance must be allowed all around so that the sleeve can adapt itself to any slight movement of the rod. Such a floating sleeve, if well designed and well made, is satisfactory and requires no attention.

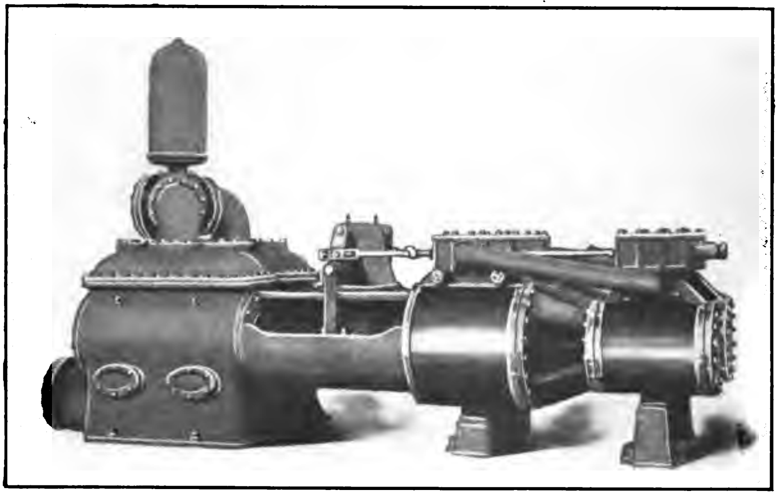


FIG. 98.—Compound pump with outboard high-pressure cylinders on support.

Improvement could be made by fitting it with some good metallic packing, but this would mean the occasional removal of several parts for inspection. Stuffing-boxes are, no doubt, the best arrangement and prevent absolutely all leakage, but they add to the length of the machine which may be prohibitive due to lack of floor space. The disadvantage of having the high-pressure cylinder attached to the back end of the low-pressure cylinder is manifest, because it means the removal of the high-pressure cylinder whenever access to the low-pressure piston is desired. On larger sizes, therefore, it is preferable to use inboard high-pressure cylinders.

64. Inboard High-pressure Cylinders.—In this arrangement the high-pressure cylinders are next to the pump end; two types have been developed, one using a single center rod, and the other a center high-pressure rod and two low-pressure rods straddling the high-pressure cylinder. These three rods are connected to a crosshead to which a plunger rod is attached. The latter type is styled a *crosshead-pattern pump* and is illustrated in Fig. 99, while the former is shown in Fig. 100.

As there is no packing between the two cylinders of the crosshead-pattern pump, they can be placed close together and some floor space is saved thereby, but at an increased initial expense.

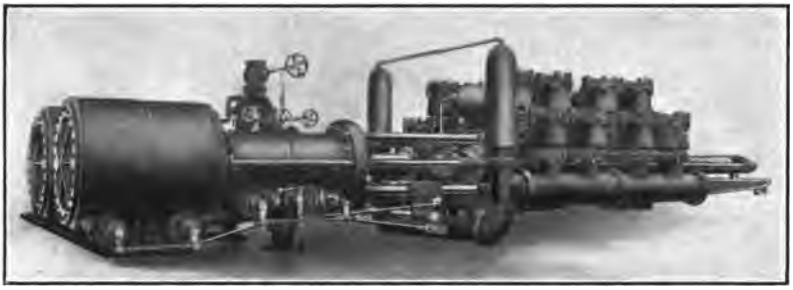


FIG. 99.—Compound pump, crosshead pattern.

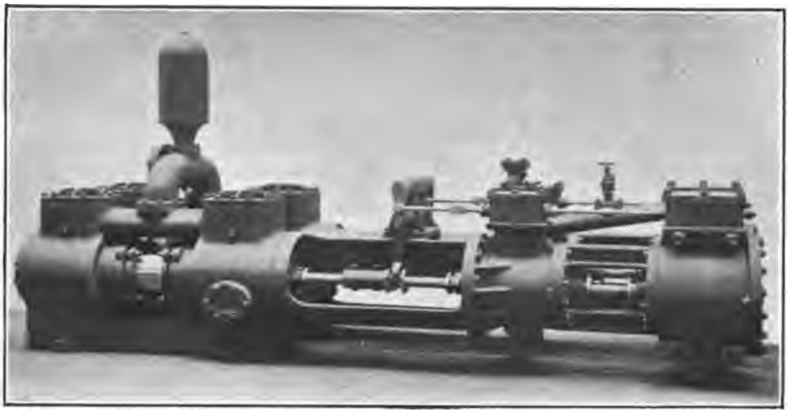


FIG. 100.—Compound pump with high-pressure cylinders inboard and center rod.

65. Cushion Ports.—Noncondensing pumps usually have the cushion ports in the low-pressure cylinders but condensing pumps should have cushion ports in both cylinders, because the cushion in the low-pressure cylinder cannot be very effective owing to the rarified condition at the point of exhaust. This is especially important in pumps having heavy reciprocating parts, such as low-service pumps, which include pressure pumps built for comparatively moderate pressures. Under "cushion" it was mentioned that for such pumps an extra long cushion should be provided and in the following some valuable rules will be given.

APPROXIMATE RULES FOR CUSHION PORTS, APPLYING TO PUMPS OF 15-INCH STROKE AND OVER

1. Simple-cylinder, Low-service Pumps.
   Use extra long cushion when diameter of water piston is equal to diameter of steam piston.
2. Simple-cylinder Pressure Pumps.
   Use extra long cushion when water pressure is 300 pounds and under.
3. Compound Noncondensing.
   Use extra long cushion on low-pressure cylinder or use regular cushion on both cylinders when the area of the water piston is equal to 1.75 times the area of the low-pressure piston.
4. Compound Condensing.
   Use regular cushion on both cylinders when the area of the water piston is 1.25 times the area of the low-pressure piston.
5. Compound Condensing.
   Use extra long cushion in both cylinders when the area of the water piston is 1.75 times the area of the low-pressure piston.
6. Compound Condensing Pressure Pumps.
   Use extra long cushion on low-pressure cylinder or regular cushion on both cylinders, on all sizes.

66. Triple-expansion Pumps.—The combined diagram of a compound direct-acting pump, Fig. 97, shows the gain in a compound pump over the simple cylinder pump. In the latter the entire expansive energy of the steam is wasted in friction by blowing through the exhaust passages into the atmosphere. In a compound pump part of this energy is saved, but there still remains an enormous loss, represented by the shaded triangle *GCD*. Efforts of pump manufacturers were, therefore, directed toward a closer approach to the economy of the cutoff engine and the next step was the development of the triple-expansion direct-acting pump, which not only gives greater economy because it utilizes a greater part of the expansive energy of the steam, but also because it is possible to use a greater number of expansions. This became desirable with the growing tendency for higher steam pressures.

The first pump of this type was brought out by the Oil City Boiler Works, now the National Transit Co., under the management of John Klein; the design was developed by A. B. Steen. Other manufacturers soon followed and this type is now considered the most suitable for small- and medium-sized waterworks. Its simplicity, reliability and durability, combined with an ability to run with very little attendance, makes it an ideal pumping engine. The comparatively low economy is more than offset by the lesser initial cost and the saving in attendance and repairs.

This type has six steam cylinders, three on each side being arranged tandem. The steam enters the high-pressure cylinder, exhausts into the first intermediate space, expands into the intermediate-pressure cylinder, exhausts into the second intermediate space, and finally expands into the low-pressure cylinder. Exactly the same effects as were shown in connection with compound pumps are obtained by the cross-exhaust valves, of which there are now two—one between the high-pressure and the intermediate cylinders and the other between the intermediate and the low-pressure cylinders. Therefore, the diagram of a triple-expansion pump may be considered as composed of three rectangles, as shown in Fig. 101, where

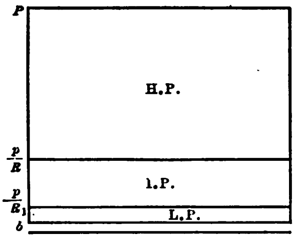


FIG. 101.—Simplified diagram of triple-expansion pump.

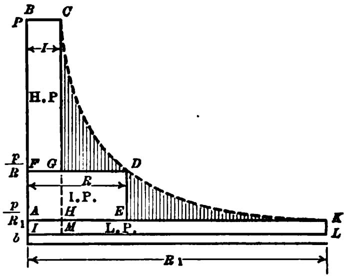


FIG. 102.—Combined diagram of triple-expansion pump.

p = Absolute initial pressure in high-pressure cylinder;

$R = \dfrac{\text{Intermediate-pressure cylinder area}}{\text{High-pressure cylinder area}};$

$R_1 = \dfrac{\text{Low-pressure cylinder area}}{\text{High-pressure cylinder area}};$

b = Back pressure.

If the first pair of cylinders be considered as forming a compound engine, then b in the formula equals $\frac{p}{R_1}$, and it was demonstrated that the efficiency is greatest for

$$R = \sqrt{\frac{p}{b}} = \sqrt{\frac{p}{\frac{p}{R_1}}} = \sqrt{R_1}$$

or

$$R^2 = R_1$$

Thus we find another simple and convenient law which says that, in a triple-expansion direct-acting pump, the ratios between the two pairs of adjacent cylinders are equal for the highest economy, or, in other words, that the best results are obtained when

$$\frac{\text{Intermediate area}}{\text{High-press. area}} = \frac{\text{Low-pres. area}}{\text{Intermediate area}} = R$$

and

$$\frac{\text{Low-press. area}}{\text{High-press. area}} = R^2 = R_1$$

Combining these three rectangular diagrams, one similar to Fig. 102 will result. Comparing this with Fig. 97, it will be observed that, for the same ratio of cylinders, the losses in the compound engine are represented by triangle *HCK* (Fig. 102), while in the triple-expansion they are represented by the two small triangles *GCD* and *EDK*.

By following the same reasoning as in the investigation of the compound pump, it will be found that the maximum economy is obtained when *HGDE* + *HKLM* is a maximum. These two rectangles have the following values:

$$HGDE = \left(\frac{p}{R} - \frac{p}{R_1}\right)(R - 1) = p - \frac{pR}{R_1} - \frac{p}{R} + \frac{p}{R_1}$$

and for $(R_1 = R^2)$ this equals

$$p - 2\frac{p}{R} + \frac{p}{R_1}$$

$$HKLM = \left(\frac{p}{R_1} - b\right)(R_1 - 1) = p - bR_1 - \frac{p}{R_1} + b$$

$$HGDE + HKLM = 2p - \frac{2p}{R} - bR_1 + b$$

$$= 2p + b - \left(bR_1 + \frac{2p}{R}\right)$$

This is a maximum when $\left(y = bR_1 + \frac{2p}{R}\right)$ is a minimum, or when

$$\frac{dy}{dR} = 2bR - \frac{2p}{R_1} = O$$

which equation is satisfied for

$$2bR = \frac{2p}{R_1}$$

or

$$R^3 = \frac{p}{b}$$

assuming again $R_1 = R^2$.

$$\frac{IP\text{ vol.}}{HP\text{ vol.}} = \frac{LP\text{ vol.}}{IP\text{ vol.}} = R = \sqrt[3]{\frac{p}{b}}$$

also

$$\frac{LP\text{ vol.}}{HP\text{ vol.}} = R^2 = R_1 = \sqrt[3]{\left(\frac{p}{b}\right)^2}$$

Let us now investigate how the work is distributed among the three cylinders. For this purpose calculate the areas of the three diagrams, in Fig. 102.

High-press. cyl....m.e.p. $= \left(p - \frac{p}{R}\right) \times 1 = p - \frac{p}{R}$

Intermediate cyl..m.e.p. $= \left(\frac{p}{R} - \frac{p}{R_1}\right)R = p - \frac{p}{R}$

Low-press. cyl....m.e.p. $= \left(\frac{p}{R_1} - b\right)R_1 = p - bR_1$

but from the previous formula,

$$R = \sqrt[3]{\left(\frac{p}{b}\right)}$$

whence,

$$bR_1 = bR^2 = \frac{p}{R}$$

which makes the low-pressure m.e.p. $= p - \frac{p}{R}$, from which it will be noted that here also the load is equally distributed among the cylinders.

The total m.e.p. is the sum of the mean effective pressures of the three cylinders, namely:

$$\begin{aligned}\text{m.e.p.} &= p - \frac{p}{R} + p - \frac{p}{R} + p - bR_1 \\ &= 3p - 2\frac{p}{R} - bR_1\end{aligned}$$

or,
$$\text{m.e.p.} = 3p - 2\frac{p}{\sqrt{R_1}} - bR_1$$

In developing these formulas it has been assumed that $R_1 = R^2$, a condition that it is safe to say will always be fulfilled, at least approximately. A slight change in the diameter of the intermediate cylinder has very little influence upon the developed power.

Table 8.—Ratios of Cylinders in Triple-expansion Pumps
b = Absolute back pressure = 16 pounds noncondensing and 5 pounds condensing.

Gage pressure in boiler	Absolute initial pressure, p	Noncondensing, R_1	Condensing, R_1
60	70	2.67	5.81
70	80	2.92	6.35
80	90	3.17	6.87
90	100	3.39	7.37
100	110	3.62	7.85
110	120	3.83	8.32
120	130	4.04	8.78
130	140	4.25	9
140	150	4.45	9
150	160	4.64	9
160	170	4.83	9
170	180	5.02	9
180	190	5.22	9
190	200	5.4	9
200	210	5.55	9

For a compound pump it was recommended not to make the ratio of cylinders over 1:4. In a triple-expansion pump a similar limit can be given of 1:3:9 above which the gain in economy is too slight to justify the first cost.

67. Designs of Triple-expansion Pumps.—Most manufacturers prefer the "telescope" design, shown in Figs. 103 and 104. The high-pressure cylinder is placed inboard, next to the pump end, and the low-pressure cylinder at the outer end.

Occasionally a pump is met where the regular compound steam end is used and a high-pressure cylinder is attached to the rear head of the low-pressure cylinder. An example of this design is shown by Fig. 105.

The design shown in Fig. 106 was developed by Henry R. Worthington in an endeavor to make each piston accessible by merely removing its own cylinder head. In this design a crosshead is used between the steam end and the pump end to which all piston and plunger rods are attached. The high-pressure piston is attached to it by one rod in the center, and the low-pressure piston by two rods on the side. These two rods straddle the intermediate cylinder, and the intermediate piston is attached to the low-pressure piston by one rod in the center. The disadvantage of this design is the great length of the low-pressure rods and the extra floor space required for removal.

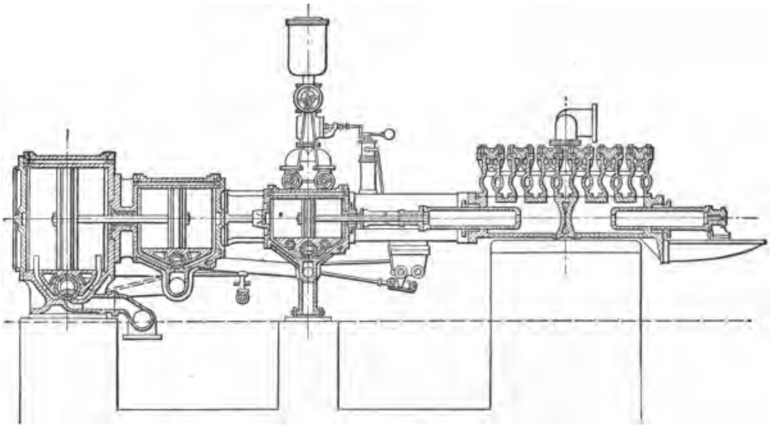


FIG. 103.—Triple-expansion pump, telescope pattern.

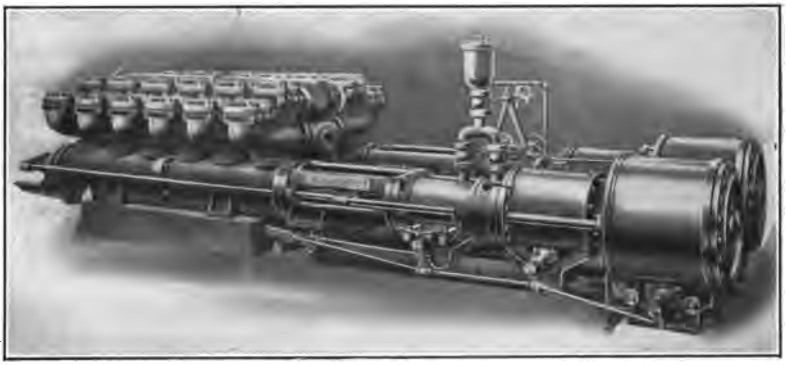


FIG. 104.—Outside view of telescope pattern.

For strokes over 24 inches, a design with a center rod is preferred. This center rod must be made in sections coupled together in the pistons, so as to make all the pistons removable without dismantling any important part of the engine.

68. Cutoff Valves.—The reciprocating parts of a triple-expansion pumping engine are necessarily very heavy and special precautions must be taken to stop the motion at the end of the stroke. For this reason the high-pressure cylinders are provided with cutoff valves, permitting the length of the stroke to be regulated to a nicety without resorting to the cross exhaust which, as explained, is a source of loss.

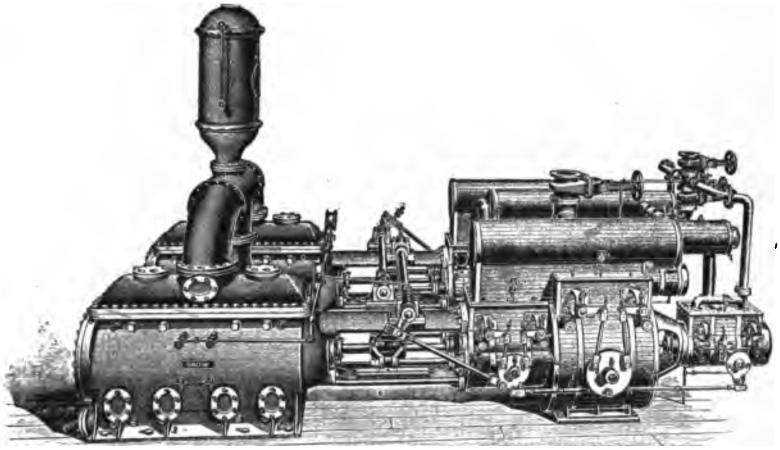


FIG. 105.—Triple-expansion pump with overhanging high-pressure cylinder.

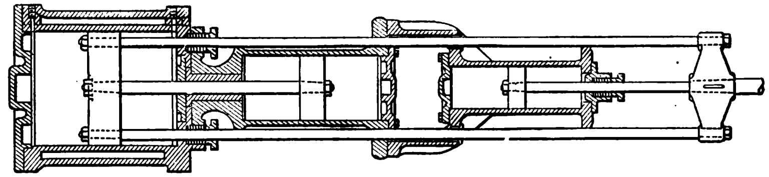


FIG. 106.—Worthington triple-expansion steam end showing arrangement of rods.

69. Cushion Valves.—As an additional safeguard, cushion valves are provided in one of the three cylinders, frequently the low-pressure; but here the steam in its tenuous condition can produce very little cushioning effect. If they are placed on the intermediate cylinder, where the exhaust steam is still at considerable pressure at the point of exhaust, this steam will offer an effective resistance to being compressed, and thus is better able to stop the motion of the pistons.

70. Steam Jackets.—Whether steam jackets contribute to the economy of an engine does not appear to be settled definitely. In direct-acting pumps, however, with their slow speed, the value of the jackets has been proven by numerous tests. The further advantage of being able to heat the cylinders before, starting and the possibility of keeping the steam end hot so as to be able to start at a moment's notice, has been an additional reason for providing the steam cylinders with jackets.

Noncondensing engines do not, as a rule, require to be jacketed, but the condensing engines should have at least the larger cylinders heated by jackets. It is usually sufficient if the high-pressure cylinder is well covered by some nonconducting material. The temperature in the jacket can never be higher than that of the incoming steam and will more likely be lower and therefore cool this cylinder at the period of admission instead of heating it. However, the opinions differ in this respect.

The jacket steam, as a rule, passes from one cylinder to the other and remains in the last, the low-pressure jacket, until it is condensed. The water of condensation is then removed by a trap. To prevent boiler pressure from accumulating in the low-pressure cylinder jacket, which is not made for high pressure, a reducing valve could be placed in the connection. However, as such a valve is not very reliable, it is better to use two traps, one to drain the intermediate jacket and the other the low-pressure jacket. A safety valve set for 50 pounds will protect the latter.

71. Reheaters.—When economy is an object, reheaters may be placed between two pairs of cylinders, making four in all. That is, on each side there will be one between the high-pressure cylinder and the intermediate cylinder and one between the intermediate and the low-pressure cylinder. These reheaters are generally cylindrical shells with heads and iron tubes heated internally by steam. The shell of the first receiver is connected to the high-pressure exhaust pipe and to the intermediate steam pipe so that the steam in passing from one cylinder to the other is heated and dried.

The first reheater is preferably heated by live steam and this same steam is then utilized in the second reheater from where it may be made to mix with the jacket steam and give up its remaining heat to the low-pressure cylinder.

The sizes of reheaters on low-duty, triple-expansion pumps are as follows: First, or high-pressure exhaust reheater—Heating surface in square inches = 0.2 to 0.3 × displacement of high-pressure in cubic feet per minute. Second, or intermediate exhaust reheater—Heating surface in square inches = 0.12 to 0.18 × displacement of intermediate cylinder in cubic feet per minute. The larger figures give higher economy than the smaller.

72. Live Steam Connections.—All triple-expansion, direct-acting pumps should be provided with an arrangement of piping and valves whereby live steam may be admitted to the intermediate cylinder for starting. These same connections will be found convenient if, on account of some unfavorable condition, the engine is short of power, or if it is desirable to run for a short time against a pressure higher than normal, as in case of fire.

If the fire pressure is only slightly higher than the domestic pressure, a small pipe connection will be found sufficient, but if very much higher, then the size of this bypass should receive careful consideration, as should also the size of the intermediate cylinders.

When the live steam connection is opened, it forms a bypass around the high-pressure cylinder and the remaining two cylinders form a compound pump and must be calculated as such. The bypass must then be proportioned to supply the intermediate cylinder but it can be kept somewhat smaller because some steam will be supplied through the high-pressure cylinder.

73. Plunger Loads.—Owing to the unavoidable mechanical friction in the engine and pump, the plunger load is only a certain percentage of the steam force, which varies from 50 per cent. in a pump of 3-inch stroke to 90 per cent. in one of 48-inch stroke. The type of the pump has also some influence upon this figure; for instance, a plunger and ring pump will not show as high a friction loss as one with packed plungers, because the latter has four large stuffing-boxes which may produce an enormous friction if not properly taken care of.

Therefore, in order to calculate the size of the steam end, it is necessary to know the mechanical efficiency (E_m); then the following formula holds good:

$$F \times E_m = L$$

and

$$F = \frac{L}{E_m}$$

where F = Steam force in pounds;
= Area of steam cylinder × m.e.p.;
L = Plunger load in pounds;
= Plunger area × water pressure in pounds per square inch.

The water pressure in this case is the total head between the two water levels, or the sum of the pressure on the discharge gage and the distance of this gage from the water level in the suction well.

Formulas for calculating the mean effective pressures have been given, and in the case of a multi-stage steam end the ratio of the cylinders must also be determined. These formulas were all developed by referring the various steam forces to the area of the high-pressure cylinder, which is contrary to the practice in calculating a steam engine, where all steam forces are referred to the area of the low-pressure cylinder. In the case of a pump, however, the high-pressure cylinder is selected because this cylinder, in the absence of a cutoff, measures out the amount of the steam that is to pass through and thus determines the power of the steam end in a greater degree than any of the remaining cylinders. If a steam engine is short of power, the power can be increased by lengthening the cutoff, but if a direct-acting pump is short of power, there is no help but to furnish a larger high-pressure cylinder.

The size of the low-pressure cylinder, on the other hand, is only of secondary importance; it has between certain limits very little influence on the steam force, and its size is often determined by considerations other than mere economy.

Therefore, an approximate ratio of cylinders will be sufficient for calculating the size of the high-pressure cylinder. Then when this has been established, the size of the low-pressure cylinder can be chosen by commercial considerations, such as initial cost, existing patterns, etc., and it is not necessary to recalculate the size for the new ratio, especially if, as is usually the case, the next larger commercial size has been selected for the high-pressure cylinder which gives additional power to cover any unfavorable conditions.

The mean effective pressure is based on the plain diagram consisting of one, two or three rectangles as in Figs. 73, 95, and 101, and the following three formulas were developed:

Simple: $$\text{m.e.p.} = p - b \qquad (1)$$

Compound: $$\text{m.e.p.} = 2p - \frac{p}{R} - bR \qquad (2)$$

Triple-Expansion: $$\text{m.e.p.} = 3p - 2\frac{p}{\sqrt{R_1}} - bR_1 \qquad (3)$$

An example will explain the application of these formulas for which will be selected a duplex outside center-packed plunger pump having two plungers 12 inches diameter and 18 inches stroke, working against a total water pressure of 100 pounds per square inch. Let the mechanical efficiency of the pump be $E_m = 78$ per cent., and assume a boiler pressure of 110 pounds.

Then, Plunger load, $L = 113.1 \times 100 = 11{,}310$ lb.

$$\text{Steam force, } F = \frac{11{,}310}{0.78} = 14{,}500 \text{ lb.}$$

The absolute initial steam pressure in the high-pressure cylinder will then be

$$p = 110 + 10 = 120 \text{ lb. (see section 59)}$$

1. Simple Cylinder Noncondensing

$p = 120$ lb. per sq. in.

$b = 16$ lb. per sq. in.

$$\text{m.e.p.} = p - b = 120 - 16 = 104 \text{ lb. per sq. in.}$$

The area of the steam cylinder will be:

$$\frac{14500}{104} = 140 \text{ sq. in. (approximately)}$$

which corresponds to a 13.35-inch cylinder. The next larger commercial size is 14 inches, so that the proper size for a simple cylinder pump will be 14 × 12 × 18 inches.

2. Compound Noncondensing

$p = 120$ lb. per sq. in.

$b = 16$ lb. per sq. in.

$$R = \sqrt{\frac{p}{b}} = \sqrt{\frac{120}{16}} = 2.74$$

$$\begin{aligned}\text{m.e.p.} &= 2p - \frac{p}{R} - bR \\ &= 2 \times 120 - \frac{120}{2.74} - 16 \times 2.74 \\ &= 152.4 \text{ lb. per sq. in.}\end{aligned}$$

referred to the area of the high-pressure cylinder.

The steam force is the same as before, namely 14,500 pounds, and the area of the high-pressure cylinder will be

$$\frac{14500}{152.4} = 95.1 \text{ sq. in.}$$

while that of the low-pressure cylinder will be

$$95.1 \times 2.74 = 260 \text{ sq. in.}$$

The corresponding diameters will then be 11 inches and 18¼ inches, and the commercial size that would be used in this pump would probably be 11 and 18 × 12 × 18 inches.

3. Compound Condensing

$$p = 120 \text{ lb. per sq. in.}$$
$$b = 6 \text{ lb. per sq. in.}$$
$$R = \sqrt{\frac{p}{b}} = \sqrt{\frac{120}{6}} = 4.47$$

The largest ratio that should be used is 4; hence $R = 4$.

Then,
$$\text{m.e.p.} = 2p - \frac{p}{R} - bR$$
$$= 240 - 30 - 24$$
$$= 186 \text{ lb. per sq. in.}$$

$$\text{Area of high-pressure cylinder} = \frac{14500}{186} = 78 \text{ sq. in.}$$
$$\text{Area of low-pressure cylinder} = 78 \times 4 = 312 \text{ sq. in.}$$

The corresponding diameters are 10 inches and 20 inches, and the commercial size would be 10 and 20 × 12 × 18 inches.

4. Triple-expansion Noncondensing

$$p = 120 \text{ lb. per sq. in.}$$
$$b = 16 \text{ lb. per sq. in.}$$
$$R_1 = \sqrt[3]{\left(\frac{p}{b}\right)^2} = \sqrt[3]{\left(\frac{120}{16}\right)^2} = 3.83$$
$$R = \sqrt{R_1} = 1.96$$
$$\text{m.e.p.} = 3p - 2\frac{p}{\sqrt{R_1}} - bR_1$$
$$= 3 \times 120 - \frac{2 \times 120}{1.96} - 16 \times 3.83$$
$$= 176.1 \text{ lb. per sq. in.}$$

$$\text{Area of high-pressure cylinder} = \frac{14500}{176.1} = 82.5 \text{ sq. in.}$$
$$\text{Area of intermediate cylinder} = 82.5 \times 1.96 = 162 \text{ sq. in.}$$
$$\text{Area of low-pressure cylinder} = 82.5 \times 3.83 = 315 \text{ sq. in.}$$

The corresponding diameters are 10.3, 14.4 and 20 inches, and the commercial size would be 11 and 14 and 20 × 12 × 18 inches.

5. Triple-expansion Condensing

$$p = 120$$
$$b = 5$$
$$R_1 = \sqrt[3]{\left(\frac{p}{b}\right)^2} = \sqrt[3]{\left(\frac{120}{5}\right)^2} = 8.32$$
$$R = \sqrt{R_1} = 2.89$$

$$
\begin{aligned}
\text{m.e.p.} &= 3p - 2\frac{p}{\sqrt{R_1}} - bR_1 \\
&= 3 \times 120 - \frac{2 \times 120}{2.89} - 5 \times 8.32 \\
&= 235.2 \text{ lb. per sq. in.}
\end{aligned}
$$

Area of high-pressure cylinder $= \frac{14500}{235.2} = 61.5$ sq. in.

Area of intermediate cylinder $= 61.5 \times 2.89 = 177$ sq. in.

Area of low-pressure cylinder $= 61.5 \times 8.32 = 515$ sq. in.

The corresponding diameters are 8.85, 15 and 25.6 inches, and the commercial size would be 9 and 15 and 26 × 12 × 18 inches.

Ratios higher than 1 : 3 : 9 should not be used.

To facilitate the selection of sizes Tables 9 to 12 will be found useful. These give the steam forces, because the plunger loads depend on the mechanical efficiency which may vary within wide limits; see Table 1. The first line (Tables 9 to 12) gives the boiler pressure in pounds per square inch, which, of course, is the gage pressure. The second line shows the proper ratios of cylinders for this boiler pressure, and the next line gives the mean effective pressures. The first column contains the diameters of the high-pressure cylinders and the figures in the body of the table represent the steam forces resulting from these conditions. No tables are given for simple-cylinder pumps, because they are easily calculated.

Applying these tables to the example which called for a steam force of 14,500 pounds we find in Table 9, in the column headed "Boiler Pressure, 110 lb.," that an 11-inch high-pressure cylinder will be good for a steam force of 14,480 pounds, which would be near enough to select an 11-inch cylinder. The ratio of cylinders is found on line 2, namely, 2.74, and the area corresponding to 11 inches diameter (95.03 square inches) multiplied by 2.74 equals 262.2 square inches, which gives a low-pressure cylinder of 18¼ inches. The size of the pump will then be 11 and 18 × 12 × 18 inches.

In Table 10, under the column headed "Boiler Pressure, 110 lb.," it will be found that a 10-inch high-pressure cylinder will give a steam force of 14,610 pounds. This would therefore be the right size. Since the ratio is 4, the low-pressure cylinder will be 20 inches diameter and the size of pump will be 10 and 20 × 12 × 18 inches.

In Table 11, for triple-expansion, noncondensing pumps it will be seen that an 11-inch high-pressure cylinder is good for 16,740 pounds and with the ratios 1.96 and 3.83 the pump size will be 11 and 14 and 20 × 12 × 18 inches.

Finally, in Table 12, for triple-expansion, condensing pumps a 9-inch high-pressure cylinder will be found to give a steam force of 14,960 pounds and with the ratios 1 : 2.88 : 8.32 the pump size will be 9, 15 and 26 × 12 × 18 inches, all of which are the same sizes as found before by calculation.

TABLE 9.—STEAM FORCES IN COMPOUND NONCONDENSING DIRECT-ACTING PUMPS

	Steam force in pounds														
Boiler press.	50	60	70	80	90	100	110	120	125	130	140	150	160	170	180
Ratio cyl's.	1.94	2.09	2.24	2.37	2.5	2.62	2.74	2.85	2.9	2.95	3.06	3.16	3.26	3.35	3.45
M.e.p.	58	73	88.5	104.1	120	136.1	152.4	168.8	177.1	185.	202	218.8	235.7	252.7	269.7
Dia. h.p. cyl. in.															
3	410	520	630	740	850	960	1,080	1,190	1,250	1,310	1,430	1,550	1,670	1,790	1,910
4	730	920	1,120	1,310	1,510	1,710	1,920	2,120	2,230	2,330	2,540	2,750	2,960	3,180	3,390
5	1,140	1,440	1,740	2,040	2,360	2,670	2,990	3,320	3,480	3,640	3,970	4,300	4,630	4,960	5,230
6	1,640	2,070	2,500	2,950	3,390	3,850	4,310	4,770	5,010	5,240	5,710	6,190	6,670	7,150	7,630
7	2,230	2,810	3,410	4,000	4,620	5,240	5,860	6,500	6,820	7,130	7,770	8,420	9,070	9,730	10,380
8	2,920	3,670	4,450	5,230	6,030	6,840	7,660	8,490	8,900	9,320	10,150	11,000	11,850	12,700	13,560
9	3,690	4,650	5,630	6,620	7,640	8,660	9,700	10,740	11,270	11,800	12,850	13,920	15,000	16,080	17,160
10	4,560	5,730	6,950	8,180	9,430	10,690	11,970	13,260	13,910	14,560	15,860	17,180	18,510	19,850	21,180
11	5,510	6,940	8,410	9,890	11,400	12,930	14,480	16,040	16,830	17,620	19,200	20,790	22,400	24,020	25,630
12	6,560	8,260	10,010	11,770	13,570	15,390	17,240	19,090	20,030	20,970	22,850	24,740	26,650	28,580	30,500
13	7,700	9,690	11,750	13,800	15,930	18,000	20,230	22,400	23,500	24,610	26,810	29,040	31,280	33,550	35,800
14	8,930	11,240	13,630	16,020	18,470	20,950	23,460	25,980	27,260	28,540	31,090	33,680	36,280	38,900	41,520
15	10,250	12,900	15,640	18,400	21,210	24,050	26,930	29,860	31,300	32,760	35,700	38,660	41,650	44,660	47,660
16	11,660	14,680	17,800	20,930	24,130	27,360	30,640	33,940	35,600	37,270	40,610	43,990	47,390	50,800	54,220
17	13,170	16,570	20,090	23,630	27,240	30,890	34,590	38,320	40,200	42,080	45,850	49,660	53,500	57,360	61,220
18	14,760	18,580	22,520	26,490	30,540	34,630	38,780	42,950	45,070	47,180	51,400	55,680	59,980	64,300	68,620
19	16,450	20,700	25,100	29,510	34,030	38,590	43,210	47,860	50,200	52,570	57,270	62,040	66,820	71,650	76,460
20	18,220	22,940	27,810	32,700	37,700	42,760	47,880	53,010	55,640	58,250	63,460	68,740	74,040	79,400	84,720
21	20,190	25,290	30,660	36,050	41,570	47,140	52,780	58,460	61,340	64,220	69,960	75,780	81,630	87,530	93,410
22	22,050	27,750	33,650	39,570	45,620	51,720	57,920	64,160	67,300	70,480	76,780	83,160	89,600	96,060	102,500
23	24,100	30,340	36,780	43,250	49,870	56,540	63,320	70,140	73,580	77,040	83,920	90,900	97,930	105,000	112,000
24	26,240	33,030	40,050	47,100	54,300	61,560	68,940	76,360	80,120	83,870	91,380	98,980	106,600	114,300	122,000
25	28,480	35,840	43,450	51,100	58,920	66,800	74,800	82,860	86,940	91,000	99,160	107,400	115,700	124,000	132,400
26	30,800	38,760	47,000	55,260	63,720	72,260	80,900	89,620	94,020	98,440	107,400	116,200	125,100	134,200	143,200
27	33,210	41,800	50,680	59,600	68,720	77,920	87,260	96,650	101,400	106,150	115,650	125,300	135,000	144,700	154,400
28	35,720	44,960	54,500	64,100	73,900	83,800	93,840	104,000	109,100	114,150	124,400	134,700	145,100	155,600	166,000
29	38,320	48,230	58,470	68,760	79,270	89,900	100,670	111,500	117,000	122,450	133,400	144,500	155,700	166,900	178,100
30	41,000	51,600	62,570	73,600	84,830	96,200	107,700	119,300	125,200	131,000	142,800	154,650	166,600	178,600	190,600

TABLE 10.—STEAM FORCES IN COMPOUND CONDENSING DIRECT-ACTING PUMPS

	Steam force in pounds														
Boiler press.	50	60	70	80	90	100	110	120	125	130	140	150	160	170	180
Ratio cyl's.	3.16	3.41	3.65	3.87	4	4	4	4	4	4	4	4	4	4	4
M.e.p.	82.1	99	116.2	133.5	151	168.5	186	203.5	212.25	221	238.5	256	273.5	291	308.5
Dia. h.p. cyl., in.															
3	580	700	820	940	1,070	1,190	1,320	1,440	1,500	1,560	1,690	1,810	1,930	2,080	2,180
4	1,030	1,250	1,460	1,680	1,900	2,120	2,340	2,560	2,670	2,780	3,000	3,220	3,440	3,660	3,880
5	1,610	1,950	2,280	2,620	2,970	3,310	3,650	4,000	4,170	4,340	4,680	5,030	5,370	5,710	6,060
6	2,320	2,800	3,290	3,780	4,270	4,770	5,260	5,750	6,000	6,250	6,740	7,240	7,730	8,230	8,720
7	3,160	3,810	4,470	5,140	5,810	6,490	7,160	7,830	8,170	8,510	9,180	9,850	10,530	11,200	11,870
8	4,130	4,980	5,840	6,710	7,590	8,470	9,350	10,230	10,670	11,110	11,990	12,870	13,750	14,630	15,510
9	5,220	6,300	7,390	8,500	9,610	10,720	11,830	12,950	13,500	14,060	15,170	16,290	17,400	18,510	19,630
10	6,450	7,780	9,130	10,490	11,860	13,230	14,610	15,980	16,670	17,360	18,730	20,100	21,480	22,850	24,230
11	7,800	9,410	11,040	12,690	14,350	16,010	17,680	19,340	20,170	21,000	22,670	24,330	25,990	27,650	29,320
12	9,290	11,200	13,140	15,100	17,080	19,060	21,035	23,010	24,000	25,000	26,970	28,950	30,930	32,910	34,890
13	10,900	13,140	15,420	17,720	20,040	22,360	24,690	27,010	28,170	29,330	31,650	33,980	36,300	38,620	40,950
14	12,640	15,240	17,890	20,550	23,240	25,940	28,630	31,330	32,670	34,020	36,710	39,400	42,100	44,800	47,490
15	14,510	17,500	20,540	23,590	26,680	29,780	32,870	35,960	37,510	39,060	42,150	45,240	48,330	51,420	54,520
16	16,510	19,900	23,360	26,840	30,360	33,880	37,400	40,910	42,680	44,430	47,950	51,470	54,990	58,500	62,020
17	18,640	22,470	26,370	30,300	34,270	38,250	42,220	46,190	48,180	50,260	54,140	58,100	62,080	66,050	70,020
18	20,890	25,190	29,570	33,980	38,420	43,010	47,330	51,780	54,020	56,240	60,700	65,140	69,600	74,040	78,500
19	23,280	28,070	32,950	37,850	42,810	47,770	52,740	57,700	60,180	62,660	67,620	72,580	77,540	82,500	87,470
20	25,800	31,100	36,500	41,950	47,440	52,940	58,440	63,940	66,680	69,440	74,920	80,420	85,920	91,420	96,920
21	28,440	34,290	40,250	46,250	52,300	58,360	64,430	70,500	73,520	76,540	82,600	88,660	94,730	100,800	106,850
22	31,210	37,630	44,110	50,750	57,400	64,050	70,700	77,360	80,680	84,000	90,660	97,300	104,000	110,600	117,250
23	34,110	41,130	48,280	55,480	62,740	70,000	77,280	84,560	88,190	91,820	99,100	106,350	113,600	120,900	128,200
24	37,140	44,790	52,560	60,400	68,300	76,220	84,140	92,060	96,020	100,000	107,900	115,800	123,700	131,600	139,600
25	40,300	48,600	57,040	65,540	74,120	82,720	91,320	99,900	104,200	108,500	117,100	125,700	134,250	142,800	151,400
26	43,590	52,560	61,700	70,900	80,170	89,460	98,760	108,000	112,700	117,300	126,600	135,900	145,200	154,500	163,800
27	47,010	56,680	66,520	76,440	86,460	96,480	106,500	116,500	121,500	126,500	136,500	146,600	156,600	166,600	176,600
28	50,560	60,960	71,550	82,200	92,980	103,750	114,500	125,300	130,700	136,100	146,850	157,600	168,400	179,200	190,000
29	54,240	65,400	76,760	88,200	99,740	111,300	122,900	134,400	140,200	146,000	157,500	169,100	180,650	192,200	203,800
30	58,040	69,980	82,140	94,380	106,700	119,100	131,500	143,800	150,000	156,200	168,600	180,900	193,300	207,700	218,000

TABLE 11.—STEAM FORCES IN TRIPLE-EXPANSION NONCONDENSING DIRECT-ACTING PUMPS

Boiler press.	70	80	90	100	110	120	125	130	140	150	160	170	180	190	200
I.p./H.p.	1.71	1.78	1.84	1.9	1.96	2.01	2.03	2.06	2.11	2.15	2.2	2.24	2.28	2.32	2.36
L.p./H.p.	2.92	3.16	3.39	3.61	3.83	4.04	4.14	4.25	4.45	4.64	4.83	5.02	5.2	5.39	5.56
M.e.p.	99.7	118.2	137.1	156.4	176.1	196	206	216.2	236.5	257.2	278	299	320.1	341.5	362.9
Dia. h.p. cyl., in.															
3	700	840	970	1,110	1,250	1,390	1,460	1,530	1,670	1,820	1,970	2,110	2,260	2,410	2,570
4	1,250	1,490	1,720	1,970	2,210	2,460	2,590	2,720	2,970	3,230	3,490	3,760	4,020	4,290	4,560
5	1,960	2,320	2,690	3,070	3,460	3,850	4,050	4,250	4,640	5,050	5,460	5,870	6,290	6,710	7,130
6	2,820	3,340	3,880	4,420	4,980	5,540	5,830	6,110	6,690	7,270	7,860	8,460	9,050	9,660	10,260
7	3,840	4,550	5,280	6,020	6,780	7,550	7,930	8,320	9,100	9,900	10,700	11,510	12,320	13,140	13,970
8	5,010	5,940	6,890	7,860	8,850	9,850	10,360	10,870	11,890	12,930	13,970	15,030	16,090	17,170	18,240
9	6,340	7,520	8,720	9,950	11,200	12,470	13,110	13,750	15,050	16,360	17,690	19,020	20,370	21,730	23,090
10	7,830	9,280	10,770	12,280	13,830	15,400	16,180	16,980	18,570	20,200	21,830	23,480	25,140	26,820	28,500
11	9,470	11,230	13,030	14,860	16,740	18,630	19,580	20,550	22,480	24,440	26,420	28,420	30,420	32,460	34,490
12	11,270	13,370	15,500	17,690	19,920	22,170	23,300	24,450	26,750	29,090	31,440	33,820	36,200	38,620	41,040
13	13,230	15,690	18,200	20,760	23,370	20,020	27,340	28,700	31,390	34,140	36,900	39,690	42,490	45,330	48,170
14	15,350	18,190	21,100	24,080	27,110	30,170	31,710	33,280	36,410	39,590	42,800	46,030	49,280	52,570	55,860
15	17,620	20,890	24,230	27,640	31,120	34,640	36,400	38,200	41,800	45,450	49,130	52,840	56,560	60,350	64,140
16	20,050	23,760	27,560	31,450	35,400	39,410	41,420	43,470	47,550	51,720	55,900	60,120	64,360	68,660	72,960
17	22,630	26,830	31,120	35,500	39,970	44,490	46,760	49,080	53,680	58,380	63,100	67,880	72,660	77,520	82,380
18	25,370	30,080	34,890	39,800	44,810	49,880	52,420	55,020	60,180	65,450	70,740	76,100	81,460	86,900	92,350
19	28,270	33,510	38,870	44,350	49,930	55,580	58,400	61,300	67,060	72,920	78,820	84,800	90,760	96,820	102,900
20	31,320	37,130	43,070	49,140	55,320	61,580	64,720	67,920	74,300	80,800	87,340	93,940	100.600	107,300	114,000
21	34,530	40,940	47,490	54,180	61,000	67,900	71,360	74,880	81,920	89,100	96,300	103,600	110,900	118,300	125,700
22	37,900	44,930	52,120	59,460	66,940	74,500	78,300	82,200	89,900	97,780	105,700	113,700	121,700	129,800	137,900
23	41,420	49,110	56,960	65,000	73,180	81,440	85,600	89,820	98,270	106,900	115,500	124,200	133,000	141,900	150,800
24	45,100	53,470	62,020	70,760	79,670	88,680	93,200	97,800	107,000	116,400	125,800	135,300	144,800	154,500	164,200
25	48,940	58,020	67,300	76,780	86,450	96,220	101,100	106,100	116,100	126,300	136,500	146,800	157,100	167,600	178,100
26	52,940	62,760	72,800	83,020	93,500	104,100	109,400	114,800	125,600	136,600	147,600	158,800	170,000	181,300	192,700
27	57,080	67,680	78,500	89,560	100,800	112,200	118,000	123,800	135,400	147,300	159,200	171,200	183,300	195,500	207,800
28	61,400	72,780	84,400	96,300	108,400	120,700	126,800	133,100	145,600	158,400	171,200	184,100	197,100	210,300	223,500
29	65,860	78,080	90,560	103,300	116,300	129,500	136,100	142,800	156,200	169,900	183,600	197,500	211,400	225,600	239,700
30	70,480	83,540	96,900	110,600	124,500	138,600	145,600	152,800	167,200	180,600	196,500	211.400	226,300	241,400	256,500

TABLE 12.—STEAM FORCES IN TRIPLE-EXPANSION CONDENSING DIRECT-ACTING PUMPS

Boiler press.	70	80	90	100	110	120	125	130	140	150	160	170	180	190	200
I.p./H.p.	2.52	2.62	2.72	2.8	2.88	2.96	3	3	3	3	3	3	3	3	3
L.p./H.p.	6.35	6.87	7.37	7.85	8.32	8.78	9	9	9	9	9	9	9	9	9
M.e.p.	144.8	167	189.5	212.2	235.2	258.4	270	281.7	305	328.3	351.7	375	398.3	421.7	445
Dia. h.p. cyl., in.															
3	1,020	1.180	1,340	1,500	1,660	1,830	1,910	1,990	2,160	2,320	2,490	2,650	2,820	2,980	3,150
4	1,820	2,100	2,380	2,670	2,960	3,250	3,400	3,540	3,830	4,130	4,420	4,710	5,010	5,300	5,600
5	2,840	3,280	3,720	4,170	4,620	5,070	5,300	5,530	5,990	6,450	6,910	7,360	7,820	8,280	8,740
6	4,100	4,720	5,360	6,000	6,650	7,310	7,640	7,970	8.260	9,280	9,950	10,600	11,260	11,920	12,580
7	5,570	6,430	7,290	8,170	9,050	9,940	10,390	10,840	11,740	12,640	13,540	14,430	15,330	16,230	17,130
8	7,280	8,400	9,530	10,670	11,820	12,990	13,570	14,160	15,330	16,500	17,680	18,850	20,020	21,200	22,370
9	9,210	10,630	12,060	13,500	14,960	16,440	17,180	17,820	19,400	20,890	22,370	23,860	25,340	26,830	28,310
10	11,370	13,120	14,880	16,660	18,470	20,300	21,200	22,120	23,950	25,790	27,620	29,450	31,280	33,120	34,950
11	13,760	15,870	18,010	20,170	22,350	24,560	25,660	26,770	28,990	31,200	33,420	35,640	37,850	40,080	42,300
12	16,380	18,890	21,430	24,000	26,600	29,220	30,530	31,860	34,500	37,130	39,770	42,410	45,050	47,700	50,330
13	19,200	22,170	25,150	28,160	31,220	34,300	35,840	37,390	40,480	43,580	46,680	49,770	52,870	55,980	59,060
14	22,290	25,710	29,170	32,660	36,210	39,780	41,560	43,370	46,950	50,540	54,140	57,740	61,320	64,920	68,500
15	25,590	29,510	33,490	37,500	41,560	45,660	47,710	49,780	53,900	58,020	62,150	66,270	70,400	74,520	78,640
16	29,110	33,580	38,100	42,670	47,290	51,960	54,280	56,640	61,320	66,000	70,700	75,400	80,080	84,800	89,480
17	32,870	37,910	43,010	48,160	58,380	58,650	61,300	63,940	69,220	74,500	79,830	85,120	90,400	95,720	101,000
18	36,850	42,500	48,220	54,000	59,860	65,760	68,700	71,690	77,600	83,540	89,500	95,440	101,350	107,300	113,250
19	41,050	47,350	53,720	60,160	66,680	73,260	76,560	79,870	86,480	93,100	99,710	106,300	113,000	119,550	126,200
20	45,490	52,460	59,540	66,660	73,900	81,180	84,820	88,500	95,820	103,150	110,500	117,800	125,100	132,500	139,800
21	50,160	57,850	65,640	73,500	81,480	89,500	93.520	97,580	105,650	113,700	121,800	129,900	138,000	146,100	154,100
22	55,040	63,480	72,040	80,660	89,420	98,220	102,620	107,100	116,000	124,800	133,700	142,550	151,400	160,300	169,200
23	60,160	69,400	78,740	88,160	97,730	107,350	112,200	117,000	126.700	136.400	146,100	155,800	165,500	175,200	184,900
24	65,500	75,560	85,730	96,000	106,400	116,900	122,150	127,400	138,000	148,500	159,100	169,700	180,200	190,800	201,300
25	71,080	82,000	93,020	104,200	115,500	126,850	132,550	138,300	149,700	161,200	172,650	184,100	195,500	207,000	218,500
26	76,880	88,670	100,600	112,650	124,900	137,200	143,350	149,600	161,900	174,300	186,700	199,100	211,500	223,900	236,300
27	82,900	95,620	108,500	121,500	134,700	148,000	154,600	161,300	174,600	188,000	201,400	214,700	228,000	241,450	254,800
28	89,160	102,850	116,700	130,700	144,800	159,100	166,300	173,500	187,800	202,200	216,600	230,900	245,300	259,700	274,000
29	95,650	110,300	125,200	140,200	155,400	170,700	178,350	186,100	201,500	216,900	232,300	247,700	263,100	27,360	294,000
30	102,400	118,000	134,000	150,000	166,300	182,650	190,850	199,100	215,600	232,100	248,600	265,100	182,500	29,820	314,600

74. High-duty Direct-acting Pumps.—The difficulties that had to be overcome in devising a steam end, that would drive a pump in a satisfactory manner without the intervention of a rotary motion, increased when an effort was made to obtain higher economy by using the steam expansively. The compound direct-acting pump and the so-called triple-expansion direct-acting pump were, in reality, only makeshifts since the steam does not expand in the cylinder, but rather in the intermediate spaces in its passage from one cylinder to the other without doing any useful work.

The following furnishes a characteristic between a low-duty and a high-duty direct-acting pump. The former uses steam the full length of the stroke; in the latter the steam is cut off at an early stage of the stroke and does useful work by virtue of its expansive force. It is not a question whether the duty is high or low. Of two engines giving the same duty, one may be a compound high-duty engine, and the other may be a triple-expansion low-duty engine; yet both may develop a duty of one hundred million foot-pounds per 1000 pounds of steam.

To utilize the expansive force of the steam, it is necessary to store up the excess energy that is developed during the first half of the stroke, and replenish it during the remainder of the stroke.

Fig. 107 shows a diagram of a cutoff engine in which steam is cut off at *B* and expands down to *C*, losing its pressure. The load remains constant throughout the stroke and may be represented by a straight line *FH*. During the first half of the stroke there is an excess of energy represented by area *FABG*, and during the remainder a deficiency represented by area *GHC*. To make such an engine run, a mechanism must be devised that will store up the work represented by the area *FABG* and will later give out the work represented by area *GHC*.

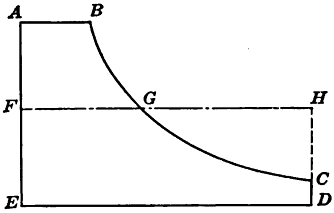


Fig. 107.—Ideal diagram of cutoff engine.

In 1881, a patent on such a devise was granted to J. D. Davies, of Covington, Ky. It consisted of a pair of oscillating cylinders fitted with rams or plungers, the outer ends of which were pivoted to a crosshead, attached to the piston rod of the engine;

see Fig. 108. This mechanism is used in the Worthington high-duty pump.

The cylinders were filled with water, or preferably oil, and connected either directly or by means of an accumulator, with the force main of the pump. By this means, a liquid pressure was maintained in these oscillating cylinders, either equal or proportional to that in the force main. Owing to the oscillating motion of the cylinders it was necessary to introduce the liquid through the trunions, as shown in Fig. 109.

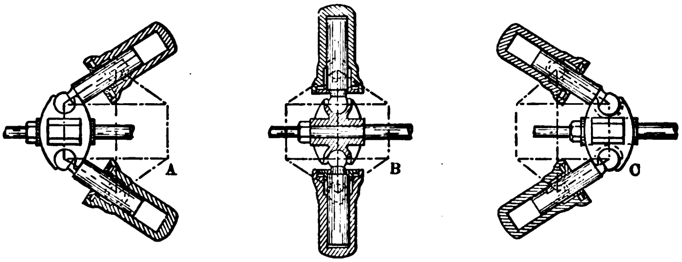


FIG. 108.—Three positions of compensating rams.

It will be noted that two compensating cylinders are placed directly opposite each other and that the vertical components of the forces exerted by the rams are thereby balanced, and only the horizontal forces remain to act on the piston rod.

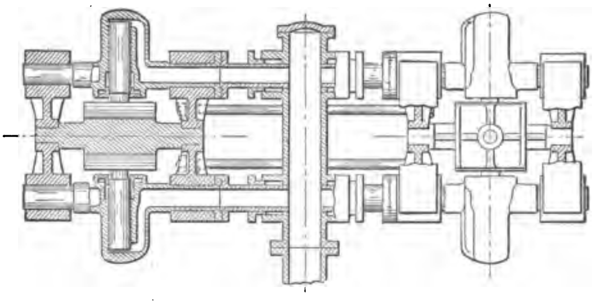


FIG. 109.—Section through compensating cylinders.

In Fig. 108 the three significant positions of the compensating rams are shown. *A* represents the beginning of the stroke where an excess of energy exists. The total energy of the steam is then utilized in overcoming the plunger load plus the compensating cylinder load. Position *B* represents the middle of the stroke where the steam has expanded down to the point *G* (Fig. 107) and where the steam force just equals the load. As this point is passed, the pressure in the compensating cylinder begins to exert itself in favor of the motion of the piston rod, and we now have: Steam force = plunger load minus compensating cylinder load, until the end of the stroke is reached.

Thus, by alternately taking up and exerting power through the difference in the angle at which their force is applied with respect to the line of motion of the plunger rod, these two cylinders perform the function of a flywheel with the important mechanical difference that they utilize the pressure in the force main instead of the energy of momentum.

75. Worthington Compound High-duty Pump.—In order to investigate the action of this compensating device properly, an ideal diagram of a compound engine with cutoffs will be drawn. A cylinder ratio of 1:4 will be assumed with cutoff in the high-pressure cylinder at 0.3 of the stroke. The theoretical point of cutoff in the low-pressure cylinder is then at one-fourth of the stroke, because the volume exhausted from the high-pressure cylinder must be equal to the volume received by the low-pressure cylinder. This would be true if the receiver were infinitely large and if there were no losses between the cylinders. This condition is shown by the dash-and-dot lines in Fig. 110. From this point of cutoff the low-pressure expansion line can be constructed and remains unchanged for any point of cutoff in the low-pressure cylinder. If, therefore, the low-pressure cutoff is changed it will move along this line.

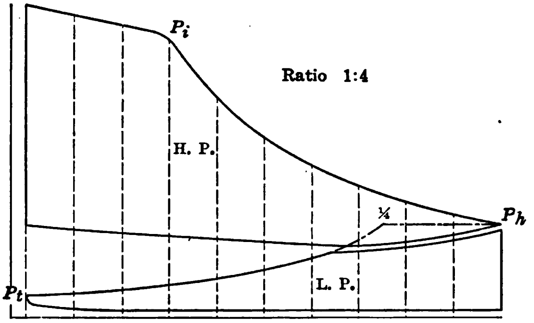


FIG. 110.—Diagram of compound steam end as taken by indicator.

A compound direct-acting pump has, as previously shown, two pairs of cylinders, one pair arranged tandem on each side. The steam distribution, therefore, differs somewhat from that in a cross-compound engine where a large receiver must be provided to form a receptacle between the two cylinders and equalize the pressure differences that would otherwise be caused by the difference in speed of the two pistons. In a tandem-compound engine the two pistons travel at the same speed and the steam can therefore exhaust from the high-pressure cylinder directly into the low-pressure cylinder, a receiver becoming unnecessary except to act as a reheater. It is then provided with heating tubes but need not be made any larger than to just accommodate the tubes. Its size being limited, the pressure between the cylinders will not be uniform and will deviate somewhat from the straight line in the diagram of a cross-compound engine.

The steam in exhausting from the high-pressure cylinder through the reheater into the low-pressure cylinder will expand along a flat expansion curve, which can be constructed by the method described under "Compound Pumps." If this line is plotted in the diagram of Fig. 110, the point of intersection with the low-pressure expansion line will determine the actual point of cutoff, which is located at about 0.33 from the beginning of the stroke. From this point on, the high-pressure piston will compress its exhaust steam into the reheater which action is shown by the back-pressure line rising, while the steam in the low-pressure cylinder is cut-off and follows its regular expansion curve. Assuming the steam to follow the law—pv = constant—and neglecting clearance, we have the following relations:

Let p_i = Abs. initial pressure at high-pressure cutoff;
p_h = Abs. terminal pressure in high-pressure cylinder;
p_t = Abs. terminal pressure in low-pressure cylinder;
$R_h = \frac{1}{\text{Cutoff}}$ = Ratio of expansion in high-pressure cylinder;
R = Ratio of cylinders;
E = Total number of expansions.

Then,

$$p_h = \frac{pi}{R_h}$$

$$p_t = \frac{p_h}{R} = \frac{p_i}{E}$$

The value of p_t should be assumed in the neighborhood of 7 pounds per square inch which determines the number of expansions, for

$$E = \frac{p_i}{7}$$

The clearance will change the above relations to some extent and its effect can easily be introduced in the computations by adding the percentage to all the volumes. Thus, if the clearance is 3 per cent., the ratio of expansion in high-pressure cylinder $= \frac{1.03}{\text{cutoff} + 0.03}$, which in the assumed case would be $\frac{1.03}{0.33} = 3.12$. The high-pressure terminal pressure is then

$$p_h = p_i \times \frac{\text{Cutoff} + 0.03}{1.03}$$

The relation between p_h and p_t is not changed if the clearances in the two cylinders are the same.

There is another difference between this engine and a Corliss crank-and-flywheel type. In the latter the throttle is always wide open, the governor adjusting the cutoff to suit the load. In the type under discussion, there is no governor, the load being practically constant. The cutoffs are adjustable individually, which is an advantage considering the independent working of the pistons. It is desirable to have some means for adjusting the stroke at any one of the four corners, and if this were attempted on all four at one time the result might prove disastrous. For this reason hand adjustment is provided for each steam valve and regulation is had by throttling the steam. This produces a falling line of admission as shown in Fig. 110.

The steam forces indicated by these two diagrams must now be added so as to obtain the force acting at every point of the stroke. In order to do this the back-pressure line must be reversed, because this line runs in the opposite direction from that of the steam line. That is, the high-pressure steam line has its beginning at the left-hand end, while the back-pressure line begins at the right, but it is evident that the effective pressure in the high-pressure cylinder at the beginning of the stroke equals the steam pressure at the beginning minus the back pressure at that point of the stroke. This condition is represented in Fig. 111, which shows all the curves in their true relation to each other.

If now the steam forces are calculated by multiplying the steam pressures by the piston areas, the corresponding forces can be added and the full line of Fig. 112 obtained. Instead of using the steam forces the same result can be obtained by dividing by the low-pressure cylinder area and then working with pressures referred to the low-pressure cylinder.

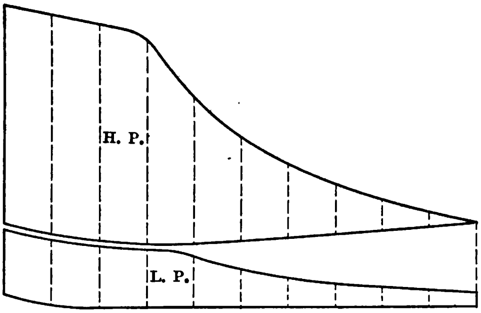


Fig. 111.—Indicator diagram with intermediate pressure lines reversed.

By drawing a horizontal line equal to the mean effective pressure, it will be found that there is again an excess of energy during the first part of the stroke and a deficiency during the latter part, which must be equalized by the compensating device. This device, however, will be partly relieved by the momentum of the reciprocating parts. At the beginning, these parts must be set in motion, which consumes part of the excess energy, whereas, at the end, they must be stopped and will give up the stored energy. This can be represented by the sloping dot-and-dash line crossing the center of the m.e.p. line. The magnitude of this inertia depends upon the speed of the pump, but since a direct-acting pump is expected to run at any speed, it is best to

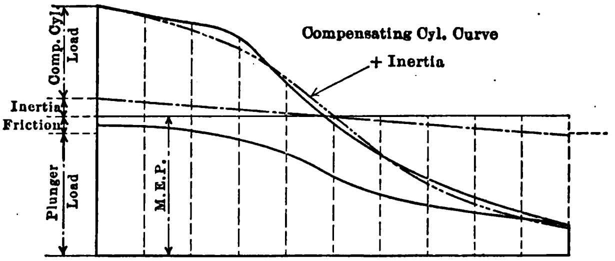


Fig. 112.—Diagram of resultant steam forces of Worthington compound high-duty pump.

calculate the compensating cylinder load for slow speed and no inertia; then reduce it by some means provided for adjustment until the stroke has assumed its proper length.

The compensating gear is represented by the diagram of Fig. 113. Point C represents the fulcrum, S the stroke, D'_1 the two diagonals, also the forces produced by the two cylinders. The total diagonal force is then $D_1 = 2D'_1$. Since O is the center of the stroke, the best results are obtained by making $OC = \frac{S}{3}$. We can then resolve each of the two diagonal forces into two components, the vertical ones which counterbalance each other and the horizontal ones $= 2H'_1 = H_1$

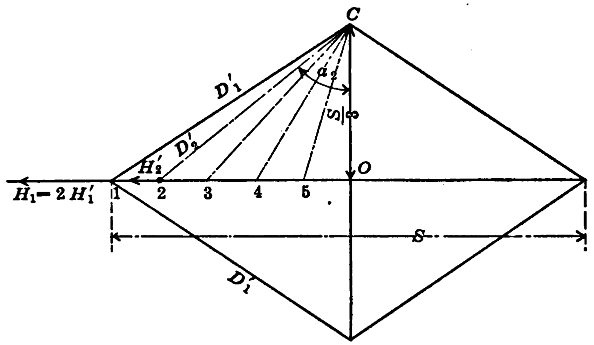


Fig. 113.—Diagram of compensating cylinder forces.

$$D'_1 = \sqrt{\left(\frac{S}{2}\right)^2 + \left(\frac{S}{3}\right)^2} = 0.6S$$

but the diagonal force is to its horizontal component as

$$D'_1 : \frac{S}{2}; \text{ hence}$$

$$\frac{H'_1}{D'_1} : \frac{S}{2 \times 0.6S} = 0.833$$

Considering both compensating cylinders we can then write:

and $$H_1 = 0.833D_1$$
$$D_1 = 1.2H_1$$

The proper values for the compensating cylinder load have been determined as follows:

Compound: $D_1 = 95$ per cent. of plunger load.
Triple-expansion: $D_1 = 75$ per cent. of plunger load.

This will make the proper values of H_1 as follows:

Compound: $H_1 = 79$ per cent. of plunger load.
Triple-expansion: $H_1 = 62\frac{1}{2}$ per cent. of plunger load.

Having determined the values at the extreme positions of the pistons, it will be easy to determine the values for the intermediate points. In order to save duplicate work it is best to determine the curve by ratios, which hold good for any force.

In Fig. 113 the first half of the stroke is divided into five parts. Selecting point 2 draw the diagonal D'_2 and designate the angle made by this line and CO by α_2; then,

$$\tan \alpha_2 = \frac{3 \times 0.4S}{S} = 1.2$$

$$\alpha_2 = 50^\circ - 12'$$

The horizontal component H_2 is then

$$H'_2 = D'_2 \sin \alpha_2$$
$$H'_2 = 0.76828\, D'_2$$

and $$H_2 = 0.76828\, D_2$$

The diagonal forces are, of course, all alike and equal to the compensating cylinder area multiplied by the oil pressure; hence $D = D_1 = D_2$, etc., and

$$H_2 = 0.76828\, D$$

Since $D_1 = 1.2\, H_1$, we have

$$H_2 = 1.2 \times 0.76828\, H_1$$
$$H_2 = 0.92\, H_1$$

In a similar manner can be found all the intermediate values as follows:

Point	1	2	3	4	5	0
H =	1	0.92	0.8	0.62	0.345	0

Following this procedure the compensating cylinder curve can be drawn in the diagram of Fig. 112.

It will be noted that the compensating line runs very close to the steam force line. The irregularities that remain are easily taken care of by the reciprocating parts which are either accelerated or retarded thereby.

If these irregularities are referred to a straight line as in Fig. 114 the propulsion curve is obtained. This corresponds to the requirements of a direct-acting pump in that it shows that the steam forces and plunger loads are nearly balanced at every point of the stroke.

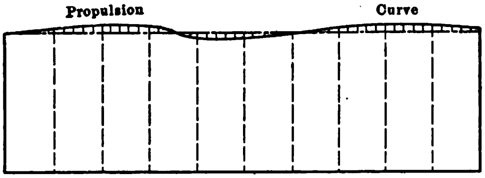


FIG. 114.—Propulsion diagram of Worthington high-duty pump.

76. Worthington Triple-expansion High-duty Pump.—The same procedure is followed in laying out the propulsion curve of a triple-expansion pump. Here three diagrams, the low-, intermediate- and high-pressure, must be added together which will produce a much smoother curve. Therefore a lighter compensating cylinder load is provided, namely, 75 per cent. diagonal and 62½ per cent. horizontal.

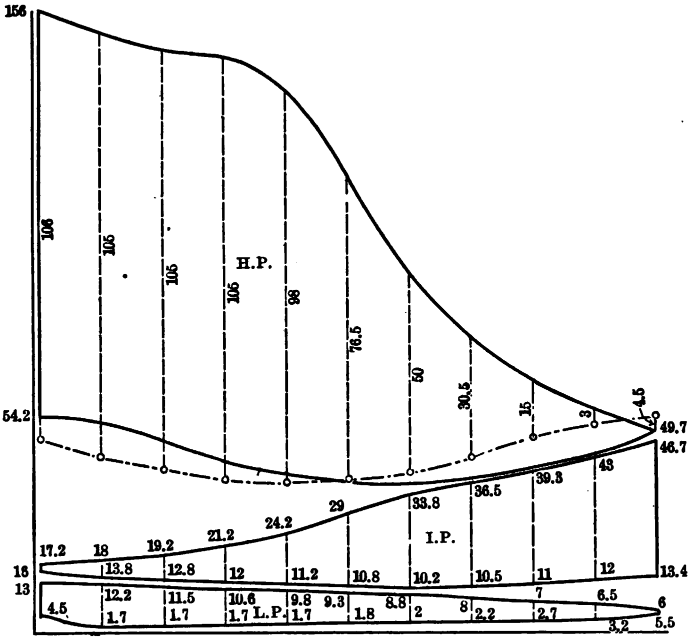


FIG. 115a.—Indicator diagram of Worthington triple-expansion high-duty pump.

Fig. 115-a shows the three actual indicator diagrams of a 27 & 42 & 76 × 45 × 36-inch vertical triple-expansion pumping engine. These were added in the manner explained and thus the combined diagram, Fig. 115*b*, was obtained. The actual compensating cylinder load was reduced to 45 per cent. of the plunger load owing to the high speed and heavy reciprocating parts (piston speed 160 feet per minute), resulting in a propulsion curve as shown in Fig. 116.

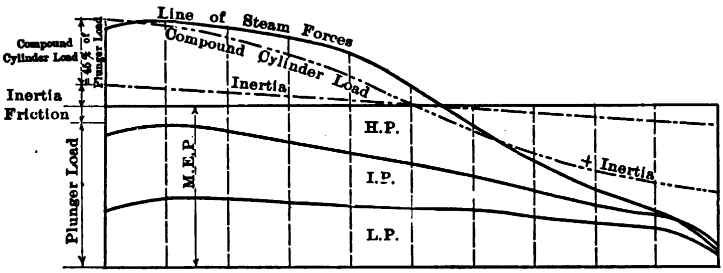


FIG. 115*b*.—Diagram of resultant steam forces of Worthington triple-expansion high-duty pump.

The compensating cylinder curve shown in Figs. 112 and 115*b* has the equation:

$$H = D \sin a$$

It is a sine curve and no mechanism that does not produce a curve of this character can be used for this purpose.

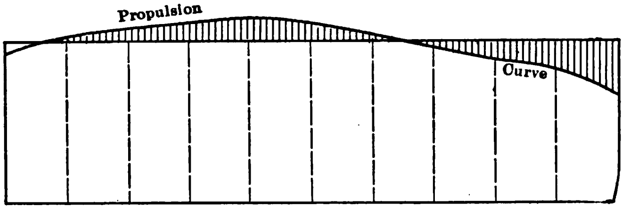


FIG. 116.—Propulsion curve of Worthington triple-expansion high-duty pump.

Fig. 117 is a sectional elevation of a triple-expansion, high-duty, direct-acting pump. The steam cylinders are fitted with regular Corliss valves, but the valve motion is necessarily different from that of a crank and flywheel engine as can be seen in the outside view of the same engine, Fig. 118. Each cylinder is equipped with a wristplate in the center which is operated from the oppo-

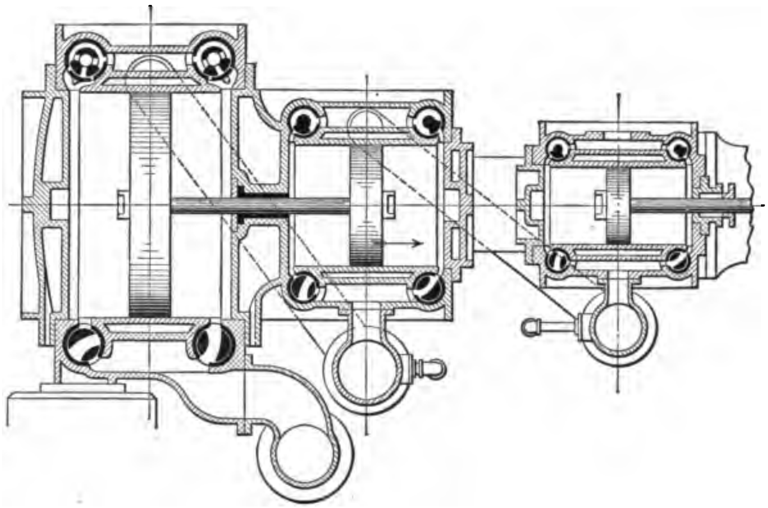


FIG. 117.—Section through high-duty triple-expansion steam end.

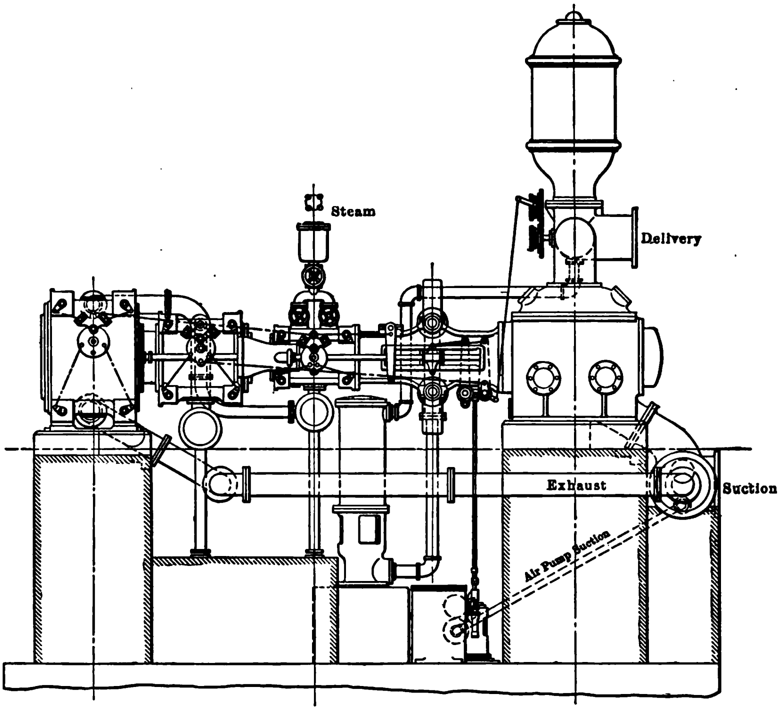


FIG. 118.—Triple-expansion, high-duty, direct-acting pump.

site side by means of suitable rock-shafts arms, and links. The exhaust valves are attached in the usual manner but the steam valves are attached to an auxiliary wristplate called the "butterfly," on account of its shape, which is pivoted to the upper part of the wristplate. This butterfly is operated from its own side and the motion thus imparted to it, combined with the motion of the wristplate, imparts to the valve the peculiar motion required for operating with a cutoff.

To study the motion, assume the opposite side to stand at the extreme end of the stroke and the side being investigated is moved forward, which motion is imparted to the butterfly and the steam valve. The steam valve is thereby closed and the point of cutoff can be determined. Then this side is supposed to stand at the extreme end of the stroke and the opposite side is moved, when it is seen that the steam valve on the other end is opening and starting off the piston.

In actual operation the motion will be modified somewhat and it is the task of the designer to design the valve motion so that at the critical points a slight motion by the opposite mechanism has no or little effect on the motion of the valve.

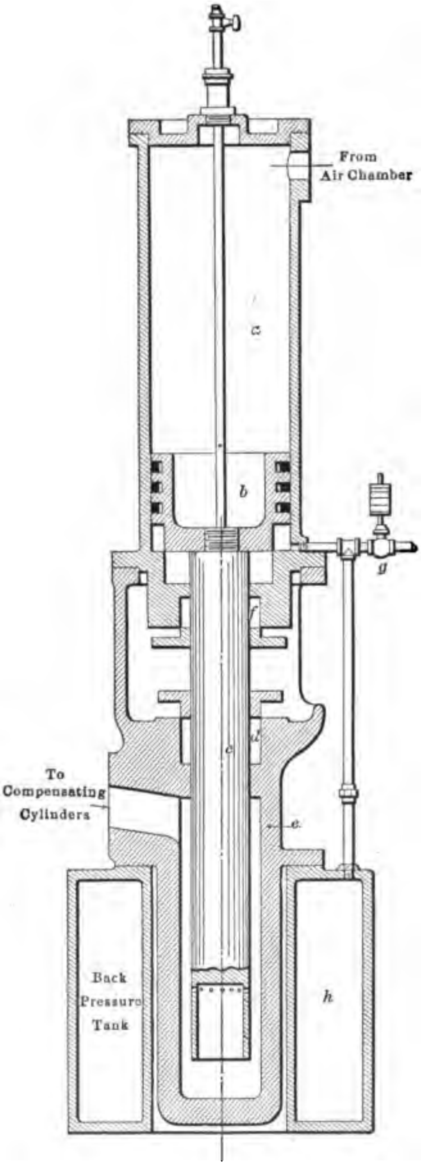


FIG. 119.—Accumulator for Worthington high-duty pump.

In order to keep the compensating cylinders small and thereby reduce the inertia of the oscillating masses, high pressures are employed, about 500 pounds per square inch. This high pressure of the liquid is produced by means of an accumulator or intensifier. This consists of an air cylinder *a*, Fig. 119, in which works an air piston *b*, attached to a ram c of much smaller diameter and extending through a stuffing-box *d* into the hydraulic cylinder *e*. The upper end of the air cylinder is connected to the air chamber of the pump while the lower end is closed against the atmosphere by the stuffing-box *f*.

The leakage past the piston will supply the lower end with air which blows off through a safety valve *g* set for any desired pressure, say 10 per cent. of the air pressure. An air tank *h* is connected to this lower part in order to insure a constant back pressure. By regulating the load on the safety valve, the back pressure can be varied, and thus the down thrust of the air piston, which produces the liquid pressure in the system, can be adjusted to a value at which the best results are obtained.

The unavoidable leakage past the air piston makes it necessary, of course, to build the air chamber extra large and to install an air compressor for the purpose of replenishing the air that is lost.

The displacement of the accumulator ram is made equal to six times the displacement of one compensating plunger. A small auxiliary pump is attached to some reciprocating part which replenishes the leakage of the liquid from the system.

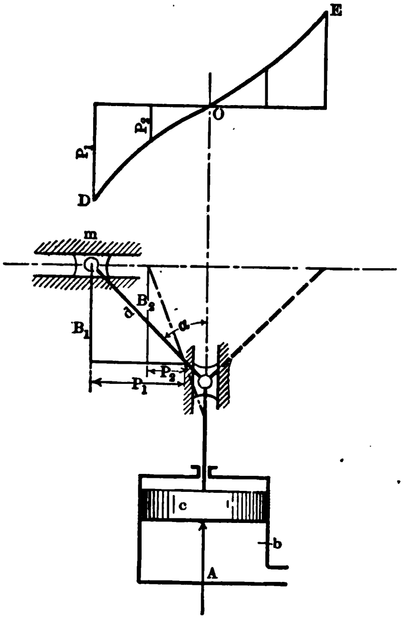


FIG. 120.—Diagram showing fallacy of constant length of link mechanism.

77. Other High-duty, Direct-acting Pumps.—Several attempts at producing a mechanism that would accomplish the same result as the compensating cylinders, previously described, were made by various manufacturers but none has met with much success.

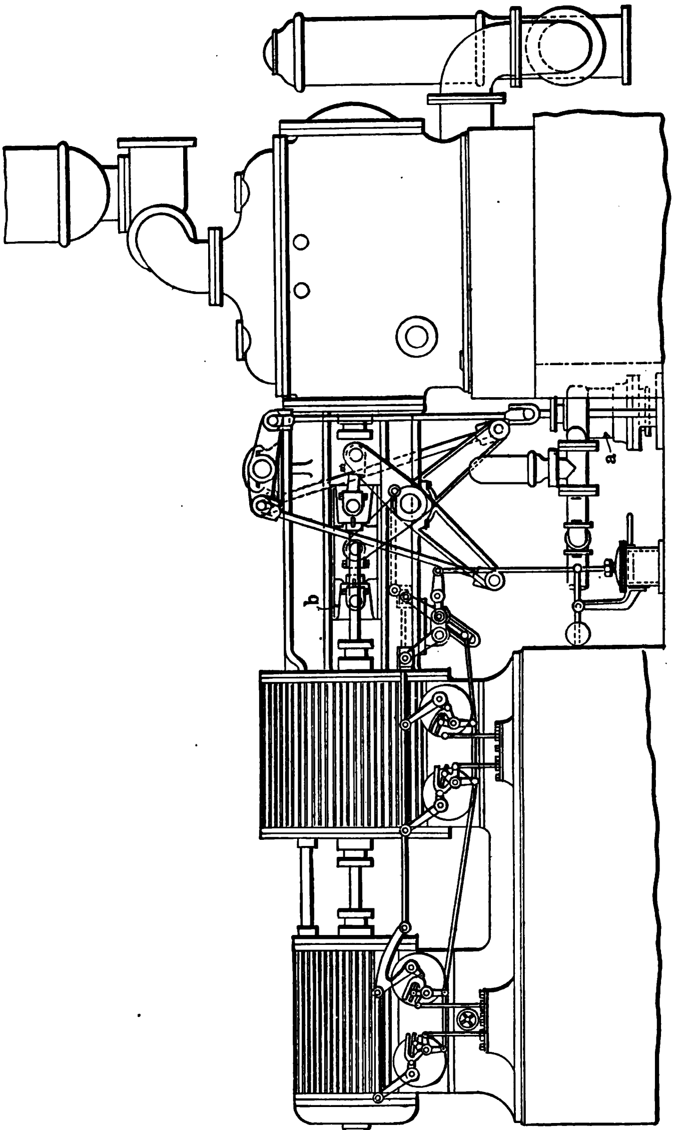


FIG. 121.—Groshon high-duty pumping engine.

The simplest form that might be suggested is a stationary compensating cylinder *b*, Fig. 120, with a piston *c* attached to the link *d*, which is guided at its ends by two crosshead guides at right angles; the horizontal crosshead is connected to the engine and pump. In this mechanism there is a constant vertical force *A* which is transmitted to the engine crosshead. This force resolves into two components, one acting at right angles to the crosshead guide, and which is destroyed thereby, and the other along the axis of the link *d*. This force can be transferred to the main crosshead *m* where it is resolved into a component B_1, which is destroyed, and a horizontal force P_1, which actuates the crosshead. If some more points, as P_2, are plotted a curve *DOE* is obtained which does not resemble the one found with the oscillating compensating cylinders. It is the line of tangents and has the equation $P = A \tan a$.

In combining the indicator cards it was seen that the resulting line of steam forces resembles very much the line of sines; therefore, it is to be expected that the mechanism shown in Fig. 120 will leave too serious irregularities in the propulsion curve. The same effect may be expected of any other mechanism employing a link of constant length. To this class belongs the following device:

78. Groshon High-duty Attachment.—Referring to Fig. 121, *a* is the stationary compensating cylinder, the force produced by it being transmitted through a series of beams and links to the engine crosshead *b*. These links, however, do not alter the effect, which is investigated in Fig. 122.

Here *A* represents the compensating cylinder force or load, as the case may be, which is transmitted through beam *H* to link *I*, to beam *J*, to link *K*, thence to the main crosshead *M*. This constant force will produce forces D_1, D_2, etc., at the point of junction between beam *J* and link *K*. The force D_1 is easily resolved into a component P_1 and one acting along the axis of *J*, which is destroyed. Since the link *K* is assumed to stand horizontally at this point, P_1 is the force acting on the main crosshead. Likewise, taking the angularity of the link into account, the force P_2 can be found and a curve *NOQ* obtained, which has the characteristic of a line of tangents.

79. D'Auria High-duty Pumping Engine.—A different principle is employed in this engine. As seen by reference to Fig. 123, an auxiliary cylinder *A* is inserted between the steam end and the pump end, and the ends of this cylinder are connected by a hydraulic loop filled with some suitable liquid. This adds consid-

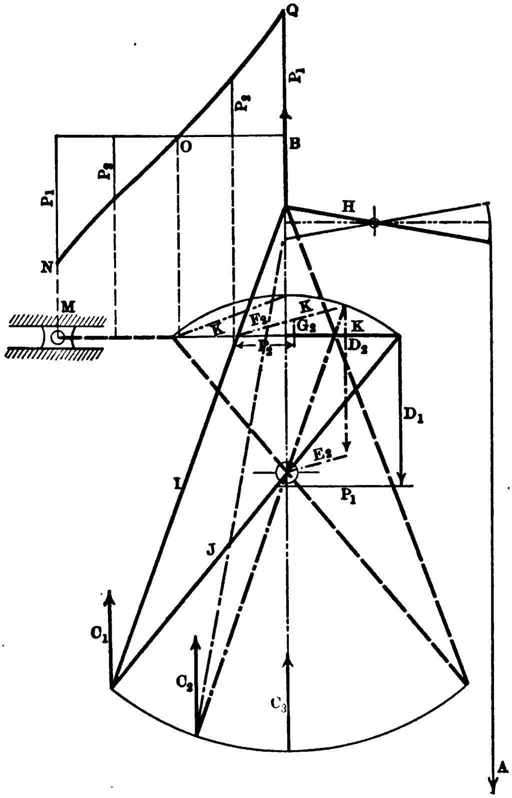


FIG. 122.—Compensating diagram of Groshon high-duty pumps.

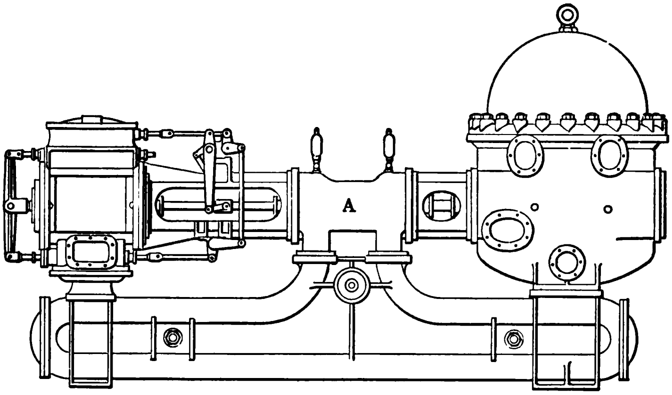


FIG. 123.—D'Auria high-duty pump.

erable weight and inertia to the reciprocating parts so that the steam can be cut off early. It seems, however, to lack the necessary flexibility demanded of a pump.

This hydraulic loop may be made right for a certain set of conditions, but would be inadequate for any other set. As has been mentioned before, a pump, as a rule, must operate under a wide range of conditions and must be able to run fast as well as at a crawling pace. Hence, any mechanism that depends upon the momentum of some heavy part, such as a flywheel or heavy water column, does not seem to be well adapted for this class of pumps and will have only a limited field.

We now come to a different principle that was developed in an endeavor to produce a direct-acting high-duty pump. In all the pumps previously described, each side was treated independently and the excess of energy developed during the first half of the stroke was stored up by some suitable mechanism and set free during the other half.

80. Heisler High-duty Pumping Engine.[1]—The designer attempted to transmit the excess of energy directly over to the other side where there was a deficiency, and for this purpose developed the mechanism shown in Fig. 124. In this, *a* and *b* are fixed fulcrums carrying two rocker arms *c* and *d*. The free ends of these rocker arms are connected by a floating link *e* and are also connected to the main crossheads *m* and *n* by means of links *f* and *g*.

To study such a device, which may be called a "cross-over motion," we will base our investigation on a resultant line of steam forces represented by the line of sines, *ABC*. If the mechanism is perfect it must produce a similar line *STU* on the opposite side. Select a point *D* of curve *ABC* and lay off the force *ED* in its proper direction, namely, *EF*. This produces a force *GH* in link *f* and the parallelogram of forces *GHIJ* gives the force *GJ* = *HI*, acting along the axis of the floating link *e*, which can be transferred to the other end and becomes *KW*. This force resolves into *KM* and *KL*; the latter is equal to the horizontal force *NO* acting on the main crosshead. In this way a curve *PQR* is obtained which does not resemble the line of sines from which the analysis was started. In fact, it is not even a symmetrical curve in that the dead center *V* does not occur until the link *e* crosses the fulcrum *b*. On the forward stroke the dead center is located in the latter half of the stroke while on the reverse stroke it is found in the first half. This mechanism was to be used on a cross-compound engine where two steam cylinders were to take the place of the four of a Worthington high-duty compound, and it was claimed by the inventor that this difference in the location of the dead center was of great advantage, because the high-pressure cylinder was cutting off early and the low-pressure cylinder late. It seems, however, that symmetry is one of the principal requirements of such a device and that the steam end should be designed for this requirement.

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[1] U. S. Patent 655232, Aug. 7, 1900.

This symmetry was the aim of the pumping engine next described.

81. DeLaval and Aborn Compensating High-duty Pumping Engine.[1]—Here two links of constant length are used and the ends are connected to the main crossheads and to an auxiliary crosshead traveling at right angles to the axis of the engine; see Fig. 125. The transmission of forces through this simple mechanism is easily traced, and, since there are links of constant length, it is not surprising to find a line of transmitted forces having the characteristics of a line of tangents, as shown by the dotted lines. This line does not approach the actual line of steam forces near enough, however, to make this mechanism a success.

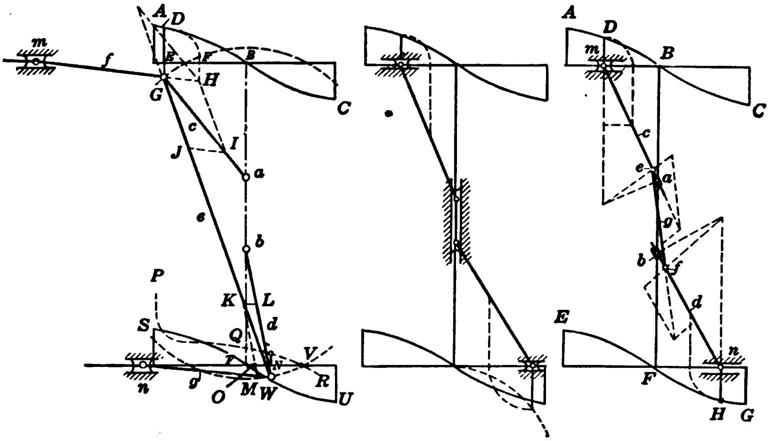

FIG. 124. C. Heisler.

FIG. 125. DeLaval and Aborn.

FIG. 126. M. Neumayer.

Figs. 124 to 126.—Cross-over motion, high-duty devices.

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[1] U. S. Patent No. 690969, Jan. 14, 1892.

82. M. Neumayer Equalizing Gear for Reciprocating Prime Movers.[1]—This seems to be a happy solution of the problem. There are two fulcrums *a* and *b*, Fig. 126, each mounting a swivel through which the two main links *c* and *d* may slide. The fulcrums correspond to the journals of the Davies compensating cylinders and the links are not of constant but of varying length, like the combination of compensating cylinder and plunger.

Two points, *e* and *f*, on these links are connected together by the center link *g* and the free ends are each connected directly to the main crossheads. The pins *e* and *f* are located so that when the crosshead *m* is in the middle of the stroke the center of the pin *e* coincides with the center of the fulcrum *a*; likewise *f* and *b*, which produces a perfectly symmetrical effect with the dead center exactly at the middle of the stroke.

Selecting point *D* of curve *ABC* and following the force *mD* through the mechanism as indicated by the dotted lines will bring us to a point *H*, which will be found located on the curve *EFG* of the opposite side. If more points are plotted they will all be found to lie on this curve which is a counterpart of the original curve *ABC*, the latter being assumed because it fairly represents the line of steam forces in a tandem-compound engine. A duplex engine of this type equipped with this mechanism should, therefore, work perfectly, and even a cross-compound is a possibility.

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[1] U. S. Patent No. 778700, Dec. 27, 1904.

CHAPTER IV

PUMP END DETAILS

The designing engineer finds himself constantly between two fires—he is expected to make every part of the machine strong beyond possibility of breaking and, on the other hand, is expected to make everything light and cheap so as to reduce the cost. Whether the machine is a standard article or a special type will determine which policy must be followed. If only a few machines are built from a certain design the saving of a few pounds of material will not justify the risk of having a part break, the replacement of which may mean an expense many times the amount saved. On the other hand, if the type is expected to be sold in quantity, it pays to build a trial pump of light design and then strengthen the parts that prove too weak.

Good cast iron has a tensile strength of 20,000 pounds per square inch, but owing to the unreliability of the average casting and to the shocks from water-hammer, it should be taken at 1500 pounds for all cast-iron parts, except in special cases where as high as 1800 pounds may be permissible.

83. Cylinders.—All curved or cylindrical surfaces may be calculated by the formula:

$$d \times p = 2t \times S$$

where d = Inner diameter of curvature in inches;
p = Internal pressure in pounds per square inch;
t = Thickness of metal in inches;
S = Allowable stress of material in pounds per square inch.

84. Spheres.—Spherical parts are twice as strong as cylinders of the same diameter, so if the radius, r, be substituted for the diameter, d, in the previous formula, we obtain

$$r \times p = 2t \times S$$

85. Flat Surfaces.—Cylinder heads and other flat surfaces are calculated by the Grash of formula:

$$t = 0.1d\sqrt{\frac{p}{200}}$$ [1]

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[1] The original Grashof formula is: $t = r\sqrt{\frac{2}{3}\frac{p}{s}}$, but if the transverse strength of cast iron be taken as 3333 pounds per square inch we obtain the simple and convenient formula given above.

This formula indicates that, for a thickness equal to one-tenth the diameter, the plate will be good for an internal pressure of 200 pounds and that the strength varies as the square of the thickness.

86. Head Bolting.—Allow 6000 pounds per square inch of the area of stud measured at the root of the thread and make the minimum diameter of the stud circle

$$B = d + 2(S + \tfrac{1}{4}\text{ in.})$$

where B = Diameter of stud circle in inches;
d = Diameter of cylinder in inches;
S = Diameter of stud in inches.

For through bolts B is determined by the outside diameter of the cylinder.

The minimum thickness of the flange is

$$f = S + \tfrac{1}{4}\text{ in.}$$

and the minimum diameter of flange

$$D = B + 2(S + \tfrac{1}{8}\text{ in.})$$

87. Pitch of Bolts.—To prevent the flange from springing away from the cylinder midway between the bolts, the pitch, or distance between the centers, should not exceed certain dimensions. It is well to make the maximum pitch (c) approximately as follows:

Steam or Water Pressure, p	Pitch, c
0 – 75 lb.	$10 \times b$
75 – 150 lb.	$7\text{-}\tfrac{1}{2} \times b$
150 – 300 lb.	$5 \times b$

where p = internal pressure in pounds per square inch;
b = diameter of bolt in inches.

No rules need be given for higher pressures, because then the bolts come very close together in any case. Special cases, where the span is relatively great, require special attention and the thickness of the flange must be calculated carefully.

88. Disk Valve.—The standard pump valve that was developed after Mr. Worthington had abolished the one large valve, and had substituted a number of small disks with the object of dissipating the shock, is a plain rubber disk such as shown in Fig. 127. Where tightness is an object and the pressure is moderate, as in air pumps, the valves are made of soft rubber; for low service up to 75 pounds they are of medium soft rubber; and for pressures up to 150 pounds medium hard rubber or bronze valves may be used. Finally, for pressures up to 300 pounds, hard rubber valves of special composition must be selected, but for this severe service special designs such as leather-faced, conical-wing or flat-seated metal valves are preferred. For pumping very hot water bronze valves or valves made of a special rubber composition must be used.

Referring to Fig. 127, the rubber disk *a* is perforated in the center and is guided on a central stem *b* which is screwed into the valve seat on a fine thread and taper. A head forming a guard for the valve is usually cast in one piece with the stem, but may, for a better class of pumps, be made separate as shown in Fig. 128. The valve is then more readily removed and the stem can be secured more safely to the seat by riveting. A spring is interposed between the guard and the valve and sometimes a brass plate protects the valve from injury caused by the action of the spring. Such a plain rubber disk may be reversed when worn on one side. The hub of the seat is attached to the rim by a number of ribs which, in the case of a rubber valve, are faced off flush with the seat and provide the necessary support. For use with a disk valve made of metal which must be ground to the seat, the ribs are depressed as shown in Fig. 129, and the valve then merely rests on the outer rim and the hub.

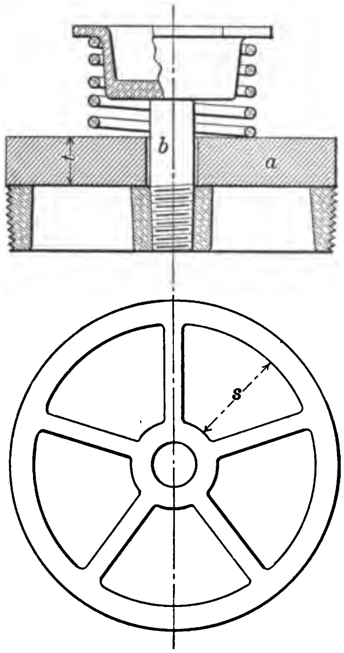

FIG. 127.—Rubber disk valve.

The thickness of the rubber valve depends upon the size of the unsupported opening and the following formula will serve as a guide. Let s be the dimension from the outside of the hub to the inside of the rim, p the pressure in pounds per square inch, and t the thickness, then $t = 0.05s\sqrt{p}$.

89. Theory of the Disk Valve.[1]—In order to properly investigate the operation of a pump valve, the pump must be reduced to the simplest elements. To this end, assume a plunger 14½ inches in diameter and 18-inch stroke, forming part of a duplex direct-acting pump and traveling at a uniform speed throughout its stroke; then coming to a stop instantly, pausing for a short time, and again resuming its velocity in the opposite direction. This condition would then be represented by a diagram such as that of Fig. 130.

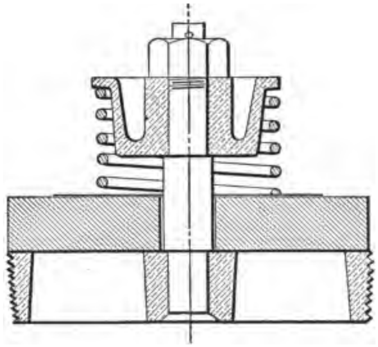

FIG. 128.—Valve with separate guard.

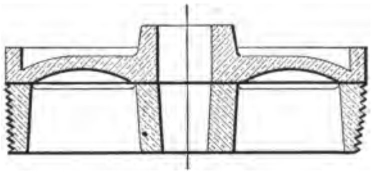

FIG. 129.—Brass disk valve.

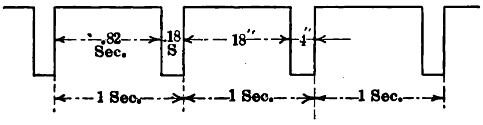

FIG. 130.—Ideal speed diagram of duplex pump plunger.

In actual operation, of course, this condition does not prevail, as the plunger cannot start from a state of rest and instantly assume a high velocity. The beginning of the curve will be more or less sloping, but its character will depend upon so many different conditions that it is impossible to tell beforehand how this acceleration will take place. At the end of the stroke the plunger is suddenly arrested in its motion by the cushion and here also a

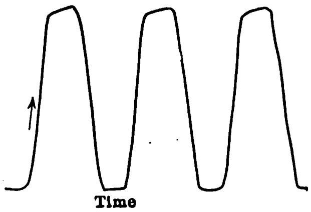

FIG. 131.—Actual speed diagram of duplex pump plunger.

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[1] O. H. Mueller, Arthur Felix, Leipzig.

certain time will elapse before it is brought to a stop. This time, however, will be so short that it may be neglected. Fig. 131 represents a diagram from a duplex pump, showing the sloping line of ascent and that of descent, and by assuming the square-cornered diagram of Fig. 130, the error will not be very great.

Assume the piston speed of the plunger to be 110 feet per minute, which equals 1320 inches per minute or 22 inches per second. If we further assume that the plunger makes one stroke of 18 inches and then rests for a length of time equivalent to a travel of 4 inches, exactly 1 second will be required for the total stroke. During

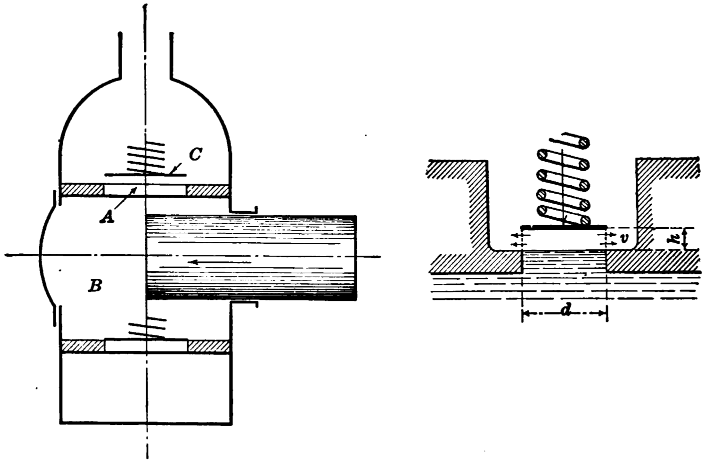


Fig. 132. Fig. 133.

Figs. 132 and 133.—Diagrammatic sketches of ideal pump valve and pump body.

the stroke the plunger displaces a certain amount of water which passes through an opening *A* in the pump body *B*, Fig. 132. The amount of water discharged is 165 sq. in. × 22 in. per sec. = 3630 cu. in. per sec. The opening is covered by a disk or valve *C*, and let this and the hole *A* both be 8 inches in diameter, neglecting the surface of the seat. At the end of the stroke, when the plunger stops, the water in the pump will cease to flow and will form a dead surface flush with the valve seat, which will represent the same condition as if the valve were located in a chamber with a solid bottom; see Fig. 133.

Consider the valve as an ideal disk without mass and let it be pressed against the seat by a spring of infinite length exerting a uniform pressure of 12 pounds. Finally, assume that, by the action of the plunger, the valve has been raised 2 inches above its seat from which position it will begin to descend as soon as the plunger has come to a stop. We will then calculate the time required for the valve to reach its seat. In air the valve would obey the law of gravitation, augmented by the spring load, that is, it would start at the moment of release with zero velocity, approach the seat with increasing velocity, and finally land on the seat with maximum velocity and a shock. Such is not the case, however, with the valve submerged in water. Fig. 134 shows that there is a certain volume of water between the valve and its seat which must be displaced before the former can reach the latter. This volume is 50 sq. in. × 2 in. = 100 cu. in. The spring load being 12 pounds and the area of the valve 50 square inches, there is a pressure in this body of water of 12 ÷ 50 = 0.24 lb. per sq. in., which is equivalent to a water column of 0.56 foot. In other words, there is a head of water, $H = 0.56$ ft., available for discharging the volume of 100 cubic inches from under the valve. The opening through which this water is discharged is the periphery of the valve multiplied by the lift h_o. This opening, at the beginning of the descent, equals

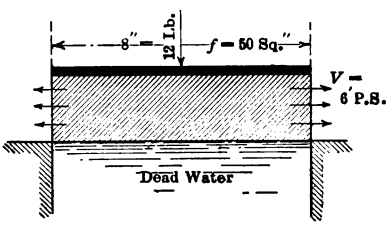


FIG. 134.—Diagram showing valve displacement.

$$\pi d h_o = 8 \times \pi \times 2 = 50 \text{ sq. in.}$$

where d is the diameter of the valve.

The case is similar to a column of water 0.56 foot high contained in a vessel with an opening at the bottom having an area of 50 square inches. According to the law of gravitation the velocity produced by this head is $v = \sqrt{2gH} = 8.02\sqrt{0.56} = 6$ ft. per sec. This means that the water will flow through the discharge opening with a velocity of 6 feet per second and since H is dependent upon the spring pressure which is constant, this velocity is also constant throughout the whole descent.

As the area of the valve happens to be the same as that of the discharge opening, obviously its velocity c, which is variable, will at the beginning be 6 feet per second. As the valve approaches the seat the size of the discharge opening will diminish, which reduces the flow and thus retards the velocity of the valve. The latter then approaches the seat, following the law:

$$h\pi dv = fc$$

where h = Lift of valve (variable);
d = Diameter of valve;
v = Velocity of water;
f = Area of valve;
c = Velocity of valve (variable).

The left-hand term is the amount of water discharged by the spring load and the right-hand term is that displaced by the valve in descending, which action may be compared with that of a plunger displacing the water in front of it. Since $f = \pi d^2 \div 4$, we find from the above equation that

$$c = \frac{h\pi dv}{f} = \frac{4hv}{d}$$

where all the terms are constant except h and c. The velocity c is, therefore, proportional to h.

Substituting the values assumed in the example

$$c = \frac{4 \times 72}{8} h = 36h \text{ in. per sec.}$$

$$= 3h \text{ ft. per sec.}$$

Taking the time into consideration, it will be noted that the time, t_o, at the beginning is zero. The distance traveled is $a = h_o - h$, and the velocity is $\dot{c} = a \div t$; or $a = ct$. Here t is the variable and a is a function of t. Hence, for an infinitesimal period of time,

$$c = \frac{da}{dt} = \frac{d(h_o - h)}{dt} = \frac{-dh}{dt}$$

but $$c = \frac{4hv}{d}$$

hence, $$\frac{4hv}{d} = \frac{-dh}{dt}$$

and, $$dt = -\frac{d}{4v} \times \frac{dh}{h}$$

from which is found the time as follows:

$$t = -\frac{d}{4v}\int\frac{dh}{h} = -\frac{d}{4v}\log_e h + k$$

The constant, K, is found by reducing this expression for t to zero for $h = h_o$ which represents the condition at the beginning of the descent. Then

$$K = \frac{d}{4v} \log_e h_o$$

and
$$t = \frac{d}{4v} (\log_e h_o - \log_e h)$$

or
$$t = \frac{d}{4v} \log_e \frac{h_o}{h}$$

An inspection of this formula reveals the fact that for $h = 0$, $t = \infty$, which means that the disk will never reach its seat. In practice, of course, the valve will come to a state of rest at a point where the film of water between the seat and valve has been reduced to such an extent that the law of gravitation does not affect it any more.

Fig. 135 shows a curve calculated according to the above formula for t. Here the different values of h were assumed and the time calculated. It will be noted that after a lapse of 0.18 second, which is the end of the pause of the plunger, the valve has come to within 0.00033 inch, and has done so without shock. It started with a high velocity of 72 inches per second, which quickly decreased and at the end of the pause, or after a lapse of 0.18 second, it was only $c = 36h = 0.00033 \times 36 = 0.012$ in. per sec., or about ¾ inch per minute.

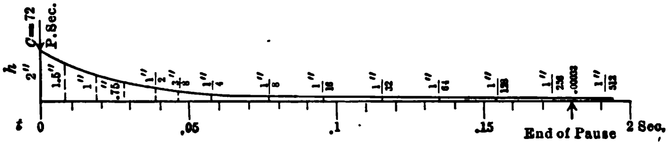


FIG. 135.—Diagram showing descent of valve during pause of duplex pump plunger.

Thus it will be seen that if the valve is given sufficient time to seat, it will do so quietly and without shock.

It must be the aim to get the valve down to its seat as quickly as possible, so as to reduce the length of the pause at the end of the stroke. The longer the plunger pauses, the faster it has to run to make up for the time lost and, as the only object of the pause is to give the valves time to seat quietly, it is obvious that we must endeavor to hasten the seating of the valve. This can be done only by increasing the initial velocity c_o.

Inspecting the formula for initial condition

$$c_o = \frac{h_o \pi d v}{f}$$

This shows that c_o can be increased by increasing v or πd. The same rules will be deduced for the crank-and-flywheel pump valve. See page 147.

In a crank-and-flywheel pump no such opportunity is afforded the valve. There is no pause at the end of the stroke, but there occurs, what appears at first sight to be just as good, a gradually diminishing speed; this, however, only partly offsets the lack of rest. The valve will in a way follow the motion of the plunger. It will gradually rise to a maximum height when the plunger has reached the middle of the stroke and then gradually descend, corresponding to the diminishing velocity of the plunger; and, if it were not for the water caught under the valve, the speed diagram of the plunger would also represent the lift of the valve. To each different speed of the plunger there corresponds a certain lift of the valve and the valve would begin to rise with a certain velocity, indicated by the slope of the lift curve, would come to rest in the middle of the stroke where the tangent to the curve is horizontal, and would descend with a gradually increasing speed to the end of the stroke when it has reached the seat. At the point of contact it would then approach the seat with a maximum velocity equal to the initial velocity; see Fig. 136.

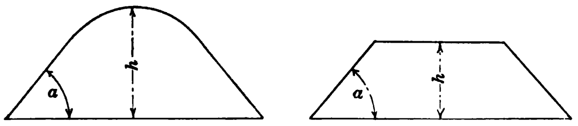


FIG. 136. FIG. 137.

FIGS. 136 and 137.—Valve lift in crank-and-flywheel pump, restricted by spring load (Fig. 136) and by a stop (Fig. 137).

It is obvious that the slope of the curve angle a increases with the maximum lift h. Therefore, if it is desired to soften the shock by decreasing the velocity at contact, it will be necessary to reduce the lift. This, however, should not be done by a stop, as this would have no effect on the action of the valve at the beginning or end and would produce a curve something like that in Fig. 137. The valve should be free to lift and find its balance, and it should be restricted by a heavy spring which produces a high velocity through the discharge opening.

The shock produced by the valve in approaching its seat with maximum velocity is enhanced by the displacement of the valve itself, because it takes time to displace the water caught under the valve and its action is therefore delayed. In other words, the valve lags behind the plunger. If the combined action of the plunger displacement and the valve displacement be analyzed, we come to a puzzling conclusion, and although this investigation does not pertain to the subject in hand, it is important that the difference in the valve action of the two great classes of pumps, the direct-acting and the crank-and-flywheel, should be understood. Besides, the action of the valve of a direct-acting pump is not represented by the diagram of Fig. 130, but by that of Fig. 131, which is intermediate between Figs. 130 and 136. Therefore, it may be said that, to a certain extent, the valve in a direct-acting pump does follow the laws laid down for the crank-and-flywheel pump.

Let s be the stroke of the pump in inches and n the number of revolutions per minute. Then the average piston speed will be

$$C = \frac{2sn}{12} \text{ ft. per min.}$$

The maximum piston speed occurs after the crank has moved through an angle of 90 deg. and is equal to the velocity of the crank pin. The crank pin moves through a distance $s\pi$ in one revolution, hence its velocity is $\frac{s\pi n}{12}$, and the maximum piston speed

$$C_{max} = \frac{s\pi n}{12} \text{ ft. per min.}$$

$$= \frac{s\pi n}{60} \text{ in. per sec.}$$

The variable velocity $C = \frac{s\pi n}{60} \sin \alpha$

Assume a pump of the following dimensions:

Diameter of plunger = $11\frac{1}{4}$ in.
Area of plunger F = 100 sq. in.
Stroke s = 22 in.
Speed n = 30 r.p.m.

The average piston speed is then

$$C = \frac{2sn}{12} = \frac{2 \times 22 \times 30}{12} = 110 \text{ ft. per min.}$$
$$= 22 \text{ in. per sec., or one stroke per second.}$$

The maximum piston speed will be

$$C_{max} = \frac{s\pi n}{60} = \frac{22 \times \pi \times 30}{60} = 34.5 \text{ in per sec. (see Fig. 138)}$$

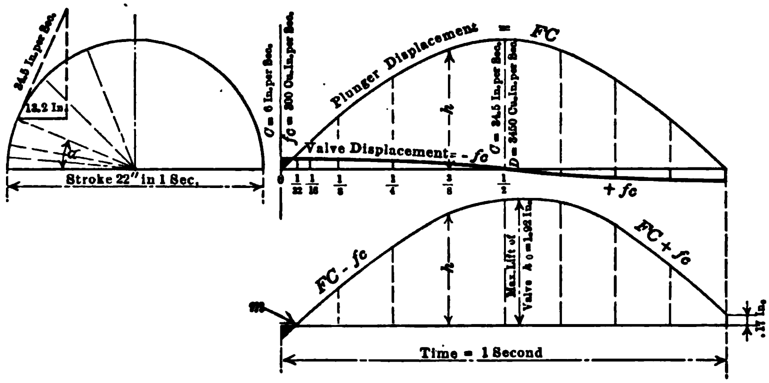


FIG. 138. FIG. 139.

FIGS. 138 and 139.—Valve action in crank-and-flywheel pump.

At the point of reversal the piston speed is zero, which shows that it is variable between the limits zero and 34.5 and its value at any point of the stroke is

$$C = 34.5 \sin \alpha$$

where α is the angle through which the crank has moved. In other words, $\alpha = 0$ when $C = 0$ and $\alpha = 90$ deg. when $C = 34.5$.

For an angle of $\frac{1}{64}$ of the semicircle or $\frac{180}{64} = 2° - 48'$ the speed is

$$C = 34.5 (\sin 2° - 48')$$
$$= 34.5 \times 0.049 = 1.68 \text{ in. per sec.}$$

Since the area of the plunger is 100 square inches the displacement of the plunger will be

$$D = 1.68 \times 100 = 168 \text{ cu. in. per sec.}$$

This quantity is flowing through the discharge area of the valve with a velocity of 72 inches per second (assuming the same valve dimensions as in the previous example) and therefore requires an opening of

$$\pi dh = \frac{168}{72} = 2.34 \text{ sq. in.}$$

As d is 8 inches,

$$h = \frac{2.34}{8 \times \pi} = 0.093 \text{ in.}$$

In other words, the valve has lifted 0.093 inches in $\frac{1}{64}$ second and its velocity c was, therefore,

$$0.093 \times 64 \doteq 6 \text{ in. per sec.}$$

In lifting this distance of 0.093 in. the valve has made room under its face for a certain amount of water, namely,

$$fc = 50 \times 6 = 300 \text{ cu. in. per sec.}$$

But as the plunger displaced only 168 cubic inches per second, it is not easy to understand what has been going on under the valve which had to lift in order to pass the plunger displacement and yet could not lift because there was not enough water to make it do so.

It was found that after the crank had moved through $\frac{1}{64}$ of the semicircle the valve had attained a velocity of 6 inches per second, and it is easy to show that this is the velocity with which the valve starts to open. To show that this is so, without resorting to differential calculus, split up angle α and calculate the valve velocity for $\frac{1}{128}$ of the semicircle. For such small angles the sines are proportional to the angles; hence everything is halved and the lift h reduces one-half in half the time, leaving the velocity the same as before, namely, 6 inches per second. Thus it will be seen that the valve starts off with the maximum velocity of 6 inches per second, reaches its maximum height where the velocity is zero, begins to descend with gradually increasing velocity and strikes the seat at maximum velocity; while in the direct-acting pumps it has been shown that the valve starts to descend with a high velocity which gradually diminishes to almost zero when the seat is reached. This fault of the crank-and-flywheel pump is augmented by the fact that the displacement of the valve itself introduces a certain distortion. It has already been pointed out that at the beginning of the stroke, for a short distance, the plunger displacement is less than the valve displacement; then there comes a point where the two are equal, after which the plunger displacement grows rapidly while the valve displacement diminishes until the middle of the stroke is reached. From this it will be seen that the amount of water flowing through the discharge opening of the valve is the difference between the two, is negative at the beginning of the stroke, then zero, and then positive until the middle of the stroke is reached, when it changes over to the sum of the two; which diminishes rapidly because the diminishing plunger displacement is so much greater than the increasing valve displacement. At the end of the stroke, where the plunger displacement is zero, the valve displacement has reached its maximum, namely, 300 cubic inches per second.

This can be shown graphically. In the upper half of Fig. 139 the plunger displacement, $D = FC$, is represented by a curve of sines. This starts off from the zero point with a certain slope representing the velocity until at the middle it has reached its maximum value, namely, $D = 3450$ cu. in. per sec. which is the product of the maximum speed ($C = 34.5$ in. per sec.) and the area of the plunger ($F = 100$ sq. in.). Since the velocity through the discharge opening of the valve is constant the lift of the valve must be proportional to the displacement D. We have then the following relation:

$$\pi dhv = D = FC$$

When α is 90 deg., the above becomes $8\pi h 72 = 3450$, whence

$$h = 1.92 \text{ in. or, in general, } h = \frac{D}{1800}$$

Therefore, the same curve may be regarded as representing the lift of the valve.

The displacement, fc, of the valve, however, produces a certain distortion. In the first half of the stroke fc must be deducted from FC, and another curve must be drawn, starting at zero with an ordinate of $d = 300$ cu. in. per sec., which gradually diminishes to zero at the middle of the stroke, then drops below the line of reference because now it must be added to FC and at the end of the stroke is again 300 cubic inches. Plotting the algebraic sum of the two, the lower curve of Fig. 139 will be had. This represents the real discharge, as well as the real lift of the valve. It will be noted that at the beginning of the stroke the lift is negative and that the valve does not begin to move until the point m is reached, where the curve crosses the line of reference. It reaches a maximum lift at a point a little behind the middle of the stroke, then descends and at the end of the stroke it is still open 0.17 inch. The plunger now reverses, thereby removing the water from under the valve, and slams the valve down on its seat with a shock.

The law expressed by the above may be written

$$\pi dhv = FC \mp fc$$

Hence, the lift $$h = \frac{FC \mp fc}{\pi dv}$$

and the velocity of the valve $c = \frac{h}{t}$ in. per sec.

It must be the aim to reduce this valve velocity as much as possible, which can be done only by increasing πd and v, the other quantities F, C, f and t being constant.

The velocity v of the water through the discharge opening can be increased by using a heavier spring, which may be expressed by the following rule: *The spring pressure should be just as heavy as the power requirements of a pump may permit.* This spring pressure will depend upon the total water pressure and may be expressed in percentages.

APPROXIMATE RULES

1. Make the spring pressure in pounds per square inch on the discharge valve area equal to ½ to 1 per cent. of the discharge pressure; maximum = 5 lb. per sq. in.

2. Make the spring pressure on the suction valve area equal to ¼ to ½ pound per square inch, if the water is lifted by suction. If the water comes in under a head, make the spring pressure on the suction valve area equal to ½ to 1 pound per square inch.

The alternative is to increase πd but it would not do to merely increase d, as this would change f, the valve area, which is determined by other considerations and is therefore constant. Since πd is the periphery of the valve it can be enlarged by using more valves of smaller diameter. If, instead of using one valve of a diameter d, n valves of diameter d_1 are employed, then

$$\frac{n\pi d_1^2}{4} = \frac{\pi d^2}{4}$$

and $$\frac{d}{d_1} = \sqrt{n}$$

or, $$n\pi d_1 = \pi d\sqrt{n}$$

showing that the new periphery is larger than the old one and that thereby the lift h and the velocity c of the valve are reduced. These formulæ hold good only for the ideal valve assumed, where the valve area = the area of the valve.

The water velocity through the valve seat is generally taken as 222 feet per minute, which gives a valve area of 45 per cent. for 100 feet piston speed. The tendency, however, is toward greater liberality in valve area, 50 and 60 per cent. frequently being demanded in waterworks specifications. The slightly increased first cost is amply repaid by good service and durability. In a pump having a 12 × 18-inch plunger,

45 per cent. for 100 ft. piston speed = 50 per cent. for 110 ft. piston speed;
50 per cent. for 100 ft. piston speed = 55 per cent. for 110 ft. piston speed;
60 per cent. for 100 ft. piston speed = 66 per cent. for 110 ft. piston speed.

and the various valve areas would be:

50 per cent. of 113 = 56.5 sq. in.
55 per cent. of 113 = 62 sq. in.
66 per cent. of 113 = 75 sq. in.

The largest diameter of a good serviceable valve in this country is considered to be 4½ inches. Assume then that it is desired to equip this pump with 4-inch valves, each giving 7.25 sq. in.; we would have 8, 9, or 11 valves.

If the pump is to deliver water against 200 pounds, the pressure necessary to open the valve can be taken at 1 pound per square inch. One pound is equivalent to 2.3 feet which gives a discharge velocity of 12.3 feet per second or 740 feet per minute. This compared with the average piston speed of 110 feet per minute gives $^{110}/_{740} = 15$ per cent., which means that the discharge opening should be 15 per cent. of the area of the plunger

$$113 \text{ sq. in.} \times 15 \text{ per cent.} = 17 \text{ sq. in.}$$

With nine 4-inch valves the total periphery will be $9 \times 4 \times \pi = 113$ in. and the lift of the valve $^{17}/_{113} \doteq 0.15$ in., say $^{3}/_{16}$ in. assuming a coefficient of contraction of 0.5, the lift will be $^{3}/_{8}$ in.

The valve should not be limited to this lift by a stop, but should be free to lift higher to adapt itself to the varying piston speed, which, as shown, is not uniform throughout the stroke. The spring pressure should be 1 pound per square inch, which on a valve of 4-inch diameter is approximately 13 pounds at ⅜-inch lift. This is sufficiently accurate for all practical purposes. There are so many disturbing influences, such as air, the varying tension of the spring and the weight of the valves, that any attempt at greater accuracy will be futile. In fact, as a rule, each manufacturer has his standard valve equipped with a standard spring, and the tension is changed only when it is found necessary to do so by the faulty operation of the pump.

90. Conical Valves.—For pressure pumps conical valves are preferred because they can be ground to a perfect seat more readily than a flat disk valve. For the latter the discharge area was the lift, *h*, times the periphery. In the conical valve, shown diagrammatically in Fig. 140,

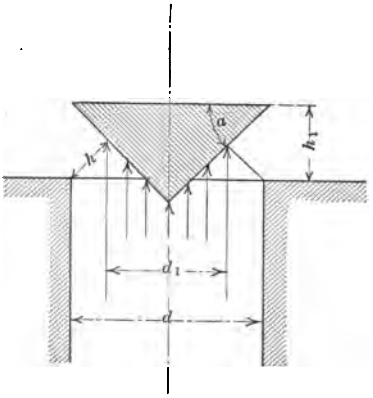


FIG. 140.—Ideal conical valve.

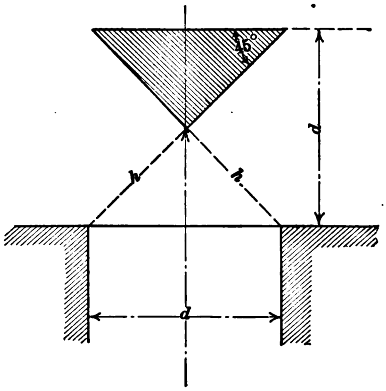


FIG. 141.—Limit of lift of conical valve.

$$h = h_1 \cos a$$

or,

$$h_1 = \frac{h}{\cos a}$$

These valves are usually made with an angle of 45 deg., which gives:

$$h_1 = \frac{h}{0.711}, \text{ or } h_1 = 1.41h$$

This means that the conical valve must lift 41 per cent. higher than the flat valve in order to discharge the same quantity of water, and it is obvious that it must close with a higher velocity. There is another peculiarity in that there is a maximum lift above which a conical valve cannot rise.

Let d, Fig. 140, be the diameter of the valve and also that of the opening in the seat. Then d_1 will be the diameter that is exposed to the lifting action of the water. If now the valve is raised to a position where the two lines h meet in a point (Fig. 141), the diameter d will be reduced to zero and the pressure produced by the spring in pounds per square inch is infinite, which makes the discharge velocity also an infinite quantity.

The discharge velocity of a disk valve is constant, but that of a conical valve increases with the lift; while the lift itself, although more than that of a disk valve at the beginning, rapidly becomes less until it has reached the value $h = d$ (Fig. 141).

This expression has, of course, no practical value. The conical valve is never made as shown, but rather as a truncated cone, and the lift is limited so as to be small compared with the diameter. However, it is true that conical valves are in every respect inferior to flat disk valves and should be used only where other considerations demand this type.

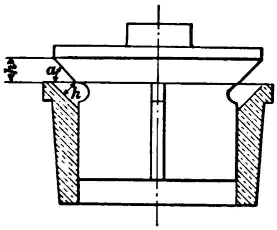


FIG. 142.—Conical wing valve.

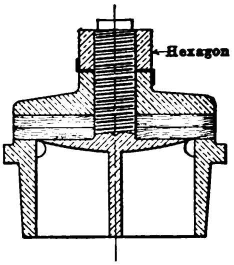


FIG. 143.—Leather-faced wing valve.

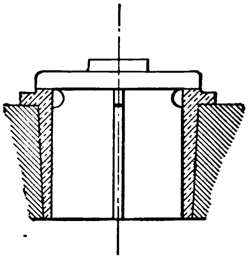


FIG. 144.—Flat-seated wing valve.

91. Wing Valves.—Fig. 142 shows the design of a conical wing valve, such as is used for pressure pumps. The under side is provided with three or four wings which guide the valve in its seat. As a rule, a valve should always be guided in its own seat and not in some separate part, such as the valve chamber cover, for it can never be expected to be tight unless guided absolutely true to the center line of the seat. A modification is the leather-faced wing valve illustrated in Fig. 143, which is used for impure mine water.

In Fig. 144 is illustrated a flat-seated wing valve which combines the good features of the disk valve with those of a wing valve. The seat can be ground to a good fit under moderately high pressures (250 pounds per square inch).

92. Clapper Valves.—For handling thick liquids, valves with large openings are required. This means that the opening through the seat must not be obstructed by any ribs or wings. For this purpose, therefore, "clapper" valves may be used (see Fig. 145).

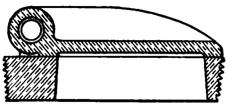


FIG. 145.—Bronze clapper valve.

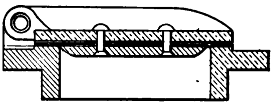


FIG. 146.—Leather-faced. clapper valve

The seat may be made round and of standard diameter so as to be interchangeable with a standard seat. The valve may be either of solid bronze or faced with leather or fiber, as in Fig. 146, the leather disks being held on by an iron binder and copper rivets. The leather should never form its own hinge as this is apt to break in a short time.

93. Ball Valves.—Another type with large unobstructed opening is the ball valve, illustrated in Fig. 147. The ball is generally of bronze, cast hollow, and the core holes are plugged. It is then turned to a true sphere and fitted to the seat. A cage is screwed over the valve to form a guide and a stop.

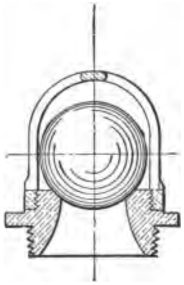


FIG. 147.—Ball valve.

For low service, such as removing the air and condensate from a condenser, the Kinghorn valve is an excellent type. This will be described under "Condenser Pumps."

An ingenious arrangement of valves is shown in Fig. 148, and is used in small pumps by the A. S. Cameron Steam Pump Works. Both valves, the suction and the discharge, work on one stem which can be withdrawn from the top. The valves can then be removed through the handholes.

94. Mechanically Operated Valves.—Crank-and-flywheel pumps equipped with mechanically operated valves were developed by Prof. A. Riedler and are successfully used abroad, especially for pumping water carrying large impurities, such as sewage where a high valve lift is desired; but no direct-acting pumps have been so equipped. In a direct-acting pump where the plunger pauses at the end of the stroke, closing of the valve could be accomplished during the pause; but since it closes without shock anyway, nothing would be gained.

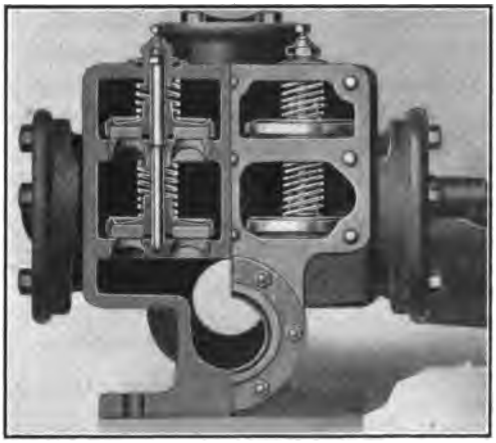


FIG. 148.—Cameron pump valve chest.

In a crank-and-fly-wheel pump the valve lags behind the plunger so that the valve must necessarily be open a certain amount at the end of the stroke. To force it down to its seat at the end of the stroke would require an infinite force and any attempt to do so results in breakage.

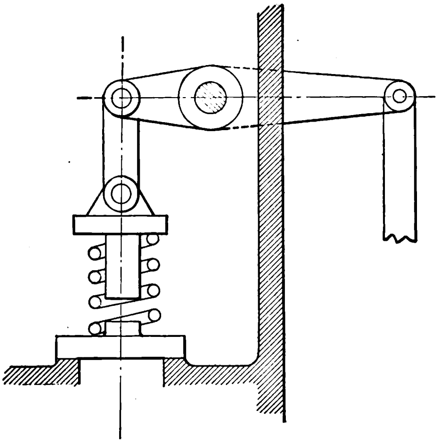


FIG. 149.—Mechanically operated valve.

To overcome the difficulty springs were inserted, as shown in the conventional sketch of Fig. 149, which is self-explanatory. The object was to provide an elastic element which would take up all shocks and prevent breakages. The real effect of this, however, was to increase the spring load at the end of the stroke when it was needed most, and release it at the beginning when it was not needed. This gave the desired effect, but at the expense of complications, consisting of levers, links, shafts, springs and eccentrics. Pumps with mechanically operated valves were developed by Prof. A. Riedler and successfully used abroad, especially for pumping water carrying large impurities, such as sewage where a high valve lift is essential.

95. Permissible Pressure on Valve Seats.—The valve seat is necessarily only a fractional part of the surface of the valve. The water pressure acting on the top of the valve will, therefore, produce a greatly increased pressure on the seat and care must be taken that this pressure does not exceed the crushing strength of the material. In Table 13 the permissible pressures are given for various materials, also the working and starting pressures.

TABLE 13.—PRESSURE ON VALVE SEATS IN POUNDS PER SQUARE INCH

Surface of seat in per cent. of free opening	Permissible pressure on seat, pounds per square inch	50 per cent.	
		Pounds per square inch	
Material		Working pressure	Starting pressure
Steel	12,000	4,000	6,000
Phos. Bronze	3,000	1,000	1,500
Gun Metal	2,100	700	1,050
Cast Iron	1,050	350	525
Leather	750	250	375
Rubber	375	125	188

When the valve is closed and in perfect contact with the seat the water pressure produced by the plunger can act only on the exposed part of the valve while the discharge pressure acts on the whole surface. The plunger must, therefore, produce a greater pressure than the discharge before the valve can open. If the surface of the seat is 19 per cent. of that of the valve, then the water pressure acts on only 81 per cent. of the area; and if the discharge pressure is 1200 pounds, then

$$1200 \times 1.00 = X \times 0.81$$

and

$$X = \frac{1200 \times 1.00}{0.81} = 1500 \text{ lb.}$$

Of course, it is not likely that this pressure is ever required to open the valve, because a pump seldom starts under full load, and if it does, it has not been standing idle long enough to let the valve come in perfect contact with the seat. After a pump has started, the film of water between the valve and the seat prevents perfect contact and only a slight increase in pressure is required to open the valve. The more valuable figures in the table are the working pressures for the various materials used in valves. The surface of the seat is given in percentage of the free opening and not of the whole surface as in the above example.

96. Valve Deck.—The valve deck is a diaphragm, cast integral with the pump body and perforated, the holes being close together so as to save valuable space. The metal between the holes is generally 1¼ inches for 4-inch valves and is reduced somewhat underneath the valve to allow the necessary clearance for the reamer and the tap. Below this recess the metal may be increased again, if desired, by reducing the opening to a circle, the area of which is a little larger than that of the valve. The former is called a plain valve deck and the latter a reinforced valve deck; see Figs. 150 and 151.

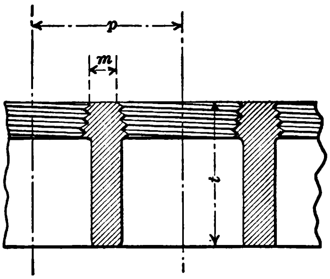


FIG. 150.—Plain pump valve deck.

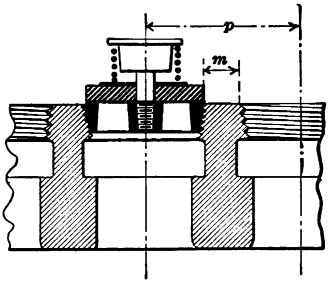


FIG. 151.—Reinforced pump valve deck.

97. Valve-deck Calculations.—First calculate the thickness, t_1, of a solid deck by the Grashof formula:

$$t_1 = d\sqrt{\frac{P}{20000}}$$

where d = Diameter of valve deck in inches;
P = Internal pressure in pounds per square inch.

If the valve deck is then considered as cut up into strips having a width equal to the pitch p (Fig. 152), the strength of each strip will be expressed by the *Modulus of Section* $= \frac{t_1^2 p}{6}$. The strength of the actual valve deck must be equal to the strength of this strip. Taking up the case of a plain valve deck (Fig. 150), it is obvious that its strength is expressed by the modulus of section of the remaining strip between the valves. This strip has a width m at the top but is somewhat reduced below the valve; therefore, the average thickness will be taken as $0.8m$ and the modulus of section is then $\frac{t^2 \times 0.8m}{6}$.

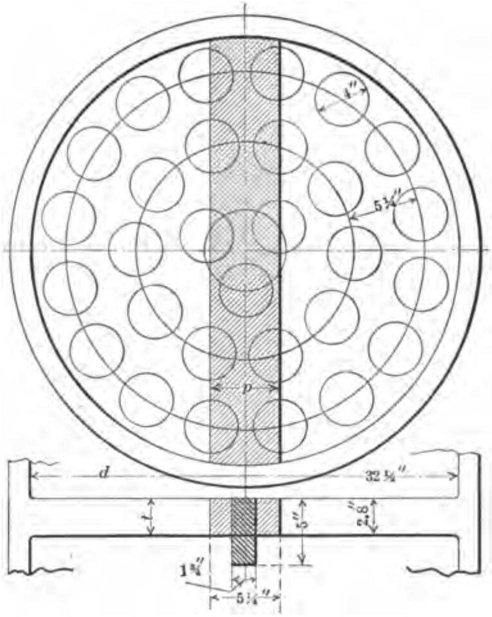


Fig. 152.—Illustrating calculation of valve deck.

To make the actual deck as strong as the calculated solid deck these two moduli must be equal; that is,

$$\frac{t_1^2 p}{6} = \frac{t^2 \times 0.8m}{6}, \text{ or } t_1^2 p = t^2 \times 0.8m$$

From Grashof's formula,

$$t_1^2 = d^2 \frac{P}{20{,}000}$$

which value substituted in above formula gives:

$$t^2 = \frac{d^2Pp}{16000m}, \text{ or } t = 0.008d\sqrt{\frac{Pp}{m}}$$

A square valve deck will have the same thickness as a round one of the same area, and since the relation of the diameter of the circle to the side of the square is 1.13 : 1, the formula becomes:

$$t = 0.009s\sqrt{\frac{Pp}{m}}.$$

where

s = Side of the square.

A rectangular valve plate of great length, where it may be assumed that no support is given lengthwise, can be considered as consisting of strips extending across the place. Each strip will then have a

$$\textit{Modulus of Section} = \frac{t_1^2p}{6}$$

where

p = Pitch of valves.

The uniformly distributed load is pwP, where w is the width of plate or length of a strip, and the bending moment

$$\frac{pwPw}{12} = \frac{t_1^2p}{6} \times S$$

(bending moment = modulus × stress), S being the allowable stress in the material.

Substituting the new modulus $\frac{t^2 \times 0.8m}{6}$ for that of the solid deck $\left(\frac{t_1^2p}{6}\right)$, we have

$$\frac{Ppw^2}{12} = \frac{t^2 \times 0.8m}{6} \times S$$

whence,

$$t^2 = \frac{6Ppw^2}{12S \times 0.8m}$$

Substituting for S, 3333 pounds per square inch

$$t = 0.013w\sqrt{\frac{Pp}{m}} \text{ (nearly)}$$

In a reinforced deck such as Fig. 151, the thickness between the valves equals m at the top, then it is reduced somewhat and at the bottom it is more than m; so that the average thickness may be taken as $1.4m$ instead of $0.8m$.

This changes the coefficients in the three formulas proportionately and we then have six formulas as follows:

Round Reinforced Deck $t = 0.006d\sqrt{\frac{Pp}{m}}$

Round Plain Deck $t = 0.008d\sqrt{\frac{Pp}{m}}$

Square Reinforced Deck $t = 0.007s\sqrt{\frac{Pp}{m}}$

Square Plain Deck $t = 0.009s\sqrt{\frac{Pp}{m}}$

Rectangular Reinforced Deck $t = 0.01w\sqrt{\frac{Pp}{m}}$

Rectangular Plain Deck $t = 0.013w\sqrt{\frac{Pp}{m}}$

Example.—Diameter of deck = 32½ in. (Fig. 152).

Water pressure, P = 150 lb. per sq. in.;
Pitch, p = 5¼ in.

Solution.—First calculate a solid deck.

$$t_1 = d\sqrt{\frac{P}{20000}}$$

$$t_1 = 32.5\sqrt{\frac{150}{20{,}000}} = 2.8 \text{ in.}$$

The valve deck may then be considered as consisting of strips 5¼ inches wide and 2.8 inches thick, and the modulus of section of each strip is given by the expression

$$\frac{2.8^2 \times 5.25}{6} = 6.9$$

The metal between the valves is m = 1¼ in. and if the deck is to be a reinforced one a uniform thickness of 1.25 in. × 1.4 = 1.75 in. may be assumed. The actual valve deck will then consist of strips 1.75 inches wide and t inches thick, and the modulus of each strip will be $\frac{t^2 \times 1.75}{6}$. This must equal that of the original strip; hence

$$\frac{t^2 \times 1.75}{6} = 6.9$$

and

$$t = 4.9 \text{ in.}$$

Applying the formula for a round reinforced deck,

$$t = 0.006d\sqrt{\frac{Pp}{m}} = 0.006 \times 32.5\sqrt{\frac{150 \times 5.25}{1.25}}$$

and

$$t = 4.9 \text{ in.}$$

which is the same as before. The valve deck would then be made 5 inches thick.

98. Pump Pistons and Plungers.—If the working part of the pump is a flat disk working in a long sleeve, it is called a "piston;" if it is a long barrel working in a short ring, either with or without packing, it is called a "plunger." Without packing the latter is spoken of as a plunger and ring pump, and with packing as an inside-packed plunger pump. A piston may be either packed or solid, and if packed it may be packed with some fibrous material or it may be provided with metallic packing rings. The plungers may also be end packed or center packed as described in articles 56 and 57.

Plunger and ring pumps should be used only for clean water because grit and dirt will rapidly destroy the fit and cause leakage. The plunger must be a good fit in the ring and should be ground to a clearance of 0.001 inch all around. The leakage will then be very slight and it can be depended upon to remain so for a long time. On account of this theoretical imperfection a plunger and ring pump is looked upon with contempt by many, who prefer a packed piston because it looks well on the drawing and is theoretically perfect. In practice, however, many difficulties are encountered in maintaining this condition. It is not easy to pack a piston just right, so that there shall be no leakage and yet be no undue friction; and succeeding in this, it will be only a short time before the packing will have worn down and the pump begun to leak.

The fact that the plunger is always traveling in a direction opposed to that of the leak assists greatly in its satisfactory working. Another great advantage is that such pumps require little attention; they can be stopped for a long time and are always ready to start the moment the throttle valve is opened.

Fig. 153 shows a packed piston of regular design. The body is provided with a recess to receive the fibrous packing, the selection of which should be left to the packing manufacturer who will recommend the best kind for the particular service. As a rule, the piston packing should be very hard and built up of layers which can be peeled off to fit the length of the packing space.

In cases where the liquid pumped must be kept clean, for such service as beer, milk, syrup or whiskey, the piston is fitted with metallic packing rings, as shown in Fig. 154; or it may be a solid block without any packing. Water pumps are generally fitted with cast-iron pistons working in bronze sleeves, which prevents the working parts from rusting together during periods of idleness. If the liquid attacks iron, as will salt water, the piston is either made wholly of bronze or is bronze lined, as illustrated in Fig. 155.

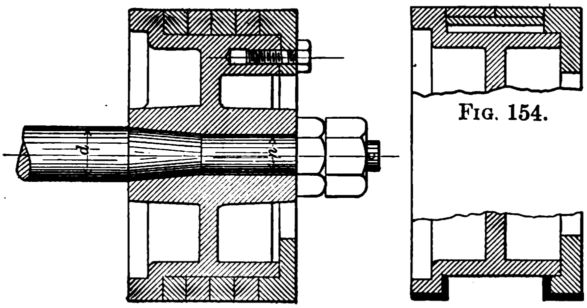


FIG. 153. FIG. 155.

FIGS. 153, 154 and 155.—Pump pistons arranged for soft packing, metallic packing, and bronze lined.

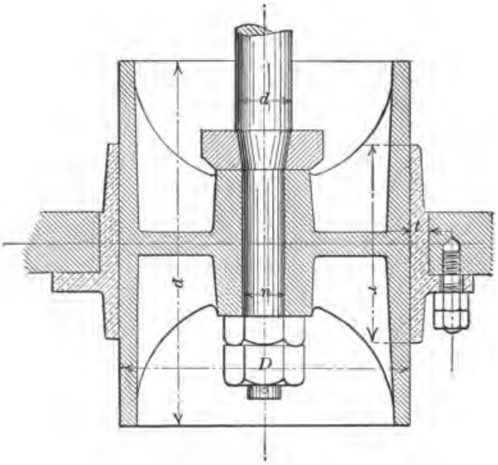


FIG. 156.—Plunger and ring.

A plunger and ring is shown in Fig. 156. In regularly fitted pumps the plunger is made of cast iron and the ring of bronze. If the plunger is made in the form of a closed barrel, in an attempt to make it a floating one, so as to relieve the weight, the extra weight of the ends and ribs will more than counteract the increased buoyancy. On small pumps the plunger ring is made of solid bronze, while on larger pumps it is of cast iron with a bronze lining inserted.

The proper lengths of plungers and rings can be determined by the following formula:

Let s = Length of stroke;
p = Length of plunger (see Fig. 156);
r = Length of ring;
D = Diameter of plunger;
$r = \frac{2}{3}s$;
$p = 1.25$ to $1.5s$.

The thickness of an all-brass ring is

$$t = 0.07\,D + \tfrac{1}{4}\text{ in.}$$

The bolting should be calculated for the full plunger load, so that if the plunger sticks in the ring, the force exerted by the steam end in tearing it loose will do no damage to the bolts.

99. Piston Rods.—These are made of steel or tobin bronze and their diameters are determined by the size of the piston nuts required. Nuts up to 1 inch have U. S. Standard threads, and those over 1 inch should have fine threads and be a snug fit on the rod to prevent jarring loose. Eight threads to the inch are recommended from 1 inch up to about 3 inches. On nuts over this size, the number of threads can be reduced somewhat.

Although a drawing may call for a tight fit, a nut cannot always be depended upon to be made to these specifications. In spite of all care, a workman may get the nut a trifle loose, and, to guard against its unscrewing and possibly wrecking the pump, it is always advisable to provide some form of locking device. While many locknuts have been devised, they are all more or less complicated and require special nuts. The simplest, best and most reliable locking device is a split or cotter-pin inserted in the rod right over the nut. An extension in the length of the rod of 3/8 inch is sufficient for a 1/4-inch pin. Should the rod wear loose in the piston, a thin metal washer placed under the nut would bring it close to the pin again.

A taper wedge, as shown in Fig. 173, is recommended by the underwriters and makes the best lock that can be devised. It is, of course, more costly and requires a specially finished hole in the rod, and a special wedge; while in the former case, the nut and cotter-pin are standard articles and can be easily procured.

The size of the piston rod nut is calculated by dividing the steam force by the allowable shearing stress, which gives the area of the rod at the bottom of the threads; from this the diameter at the bottom of the thread can be found. By adding to the diameter twice the depth of the thread, the outside diameter of the thread is obtained.

Example.—The diameter of the steam cylinder is 14 inches and the steam pressure is 150 pounds per square inch. Find the size of nut.

Solution.—Steam force = area × pressure.

$$\frac{14^2 \times \pi}{4} \times 150 = 154 \times 150 = 23{,}000 \text{ lb.}$$

For a steel rod, assume the stress to be 7000 pounds per square inch. Hence, the area at the bottom of the thread is

$$\frac{23000}{7000} = 3.3 \text{ sq. in.}$$

which is the area of a diameter of 2.05 inches.

From any table of U. S. threads, the difference between the outer diameter and that at the bottom of the thread for any number of threads per inch can be found. The following gives the differences for the numbers of threads now being dealt with:

No. of threads	Outside diameter, n, inches	Diameter at bottom of thread, inches	Difference, inches
13	½	0.400	0.1
11	⅝	0.507	0.118
10	¾	0.620	0.13
9	⅞	0.731	0.144
8	1	0.837	0.163
8	3	2.837	0.163
6	3-¼ and over	3.035	0.215

We decide to use eight threads per inch and must therefore add 0.163 to this inner diameter of 2.05 inches.

$$2.05 + 0.163 = 2.213$$

or say,

$$n = 2\tfrac{1}{4} \text{ in. diameter}$$

The rod is fastened to the piston by means of a taper which will be taken as ¾ inch to the foot on the diameter, or ⅜ inch on the side. This fine taper binds the rod and adds considerably to the reliability of the connection. This taper will be made 4 inches long. Three-fourths of an inch per foot is equivalent to 1/16 inch per inch, so the rod will taper 4 × 1/16 = ¼ in. in 4 inches, or from 2¼ inches to 2½ inches, which is then the large diameter of the taper and 2¼ inches is the diameter of the thread, or the size of the nut.

Comparing this taper with a square shoulder of equal strength, we can allow a compression stress of 30,000 pounds per square inch. Then

Area of 2½ in. diameter = 4.9 sq. in.
Area of 2¼ in. diameter = 3.97 sq. in.
Difference = 0.93 sq. in.

for a force of 23,000 pounds per square inch, or

$$\frac{23000}{0.93} = 25{,}000 \text{ lb. per sq. in. (nearly)}$$

which is below the permissible stress.

To prevent the constant hammering from driving the piston rod into the piston with the danger of bursting it, a small square shoulder is added at the point where the taper begins. This is usually made 1/16 inch but should not be less than ⅛ inch on rods 2½ inches or over. The diameter of the rod *d* (Fig. 153) will then be 2⅝ inches, which will be found ample for withstanding all compression stresses. It is not necessary to figure the rod as a long column since it is well supported in its length at various points.

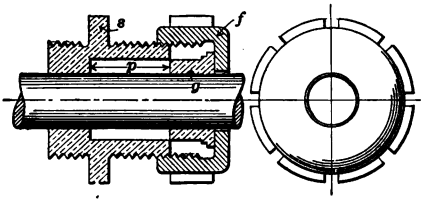


Fig. 157.—Screwed stuffing-box.

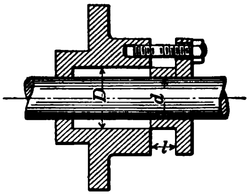


Fig. 150.—Bolted stuffing-box.

To facilitate the removal of the plunger it may have a straight fit over the rod, as shown by Fig. 156. A good substantial shoulder is then provided by means of a collar which is fitted to the rod by a taper fit. This taper is made 1½ inches in 1 foot and the collar is short, about 2 inches in length.

100. Stuffing-boxes.—The stuffing-boxes may be of various forms, two types having been well established. In the stuffing-box with a screwed gland (Fig. 157), the end *s* is provided with a thread and the follower *f* is screwed on, thereby pushing the gland *g* into the packing space *p*.

A stuffing-box with a bolted gland (Fig. 158) has no follower, but the gland itself is provided with means for adjustment. It may either have an oval or lozenge-shaped flange with two bolts, or it may be round with two or more bolts, depending upon the size. The bolts should be calculated for the maximum load that may come on the gland, namely:

$$\text{Load} = (\text{area } D - \text{area } d) \times \text{pressure.}$$

A stuffing-box should be of a depth p to accommodate six rings of packing. The shallower a box, the harder the packing must be compressed in order to make it tight, and this is liable to score the rod. The length l of the gland should be equal to two rings of the packing.

Pumps that remove liquids from a receptacle under a high vacuum should be provided with water-sealed stuffing-boxes.

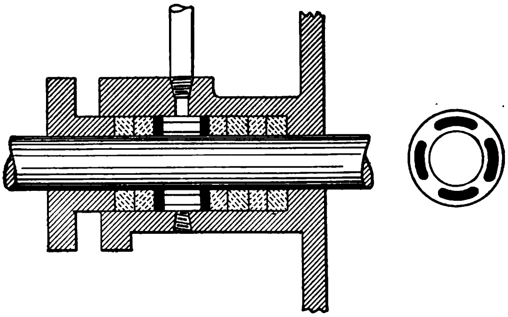


FIG. 159.—Water-sealed stuffing-box, lantern type.

An ordinary stuffing-box, provided it is long enough, can be water sealed by inserting a lantern, as shown in Fig. 159. First, four rings of packing are placed in the box, then the lantern, then two more rings, and the free space thus left by the lantern is connected to the discharge chamber of the pump. This absolutely prevents any air from leaking in, and the stuffing-box, instead of having to be tight against a high vacuum, must only be tight against a moderate water pressure. Fig. 160 shows an open water box with two stuffing-boxes, while Fig. 161 is a modification of this.

101. Handholes.—The suction chamber, pump chamber and force chamber must be provided with handholes for access to the valves, inspection and cleaning. In small pumps the back head may serve as a handhole for the outboard pump chamber, but if this head is large and inconvenient to handle, an extra opening should be provided on the side.

The size of the handhole varies with the size of the pump. In small pumps, where there is not much space, it is just large enough to get the hand and arm through and no opportunity is left for seeing what one is doing. In large pumps the handholes may assume the proportions of manholes. Attention should be given to placing the handholes properly, so that no valve is at a distance greater than can be reached by the hand conveniently. Many illustrations will be found in this book showing the location of the various handholes.

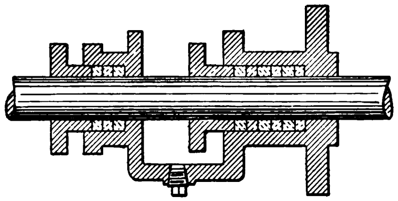


FIG. 160.—Water-box type water-sealed stuffing-boxes.

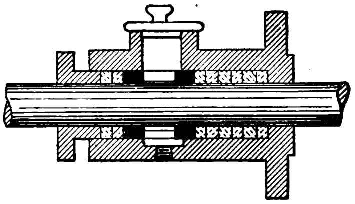


FIG. 161.—Combination box and lantern water-sealed stuffing-boxes.

102. Materials Used for Pumping Various Liquids.—The following names are in use for variously fitted pumps:

"Regularly Fitted."—This means that the pump has a cast-iron plunger or piston working in a bronze ring, lining or sleeve; that the piston rods are of steel and the valve seats, springs and plunger nuts, as well as all inside screws and nuts, are made of bronze; and that the valves are of rubber or bronze.

"Brass Fitted."—Here the rods are made of "tobin bronze." The piston is of either solid bronze or bronze lined, and the throats of the stuffing-boxes and the glands are bronze lined.

"All Iron Fitted."—Pumps for handling ammonia or tar must be built without any bronze or brass coming in contact with the substance pumped. The valve seats are then made of malleable iron, the springs are steel, the cast-iron piston works in the solid bore of the cylinder and the plunger nuts are iron.

"All Bronze."—Pumps for handling acid that attacks iron are made entirely of bronze, which in some cases must not contain any zinc.

"Lead-lined and Wood-lined Pumps."—Large pumps that would be too expensive if made of bronze may be made of iron;

the whole interior is then lined with some acid-resisting material, such as lead, wood or cement.

The following list gives some hints as to how a pump should be fitted for handling various liquids.

Ammonia	All iron
Beer	Brass fitted
Bichloride of mercury	All iron
Brine	Brass fitted
Calcium brine	Regularly fitted
Calcium chloride	All iron
Cane juice	Brass fitted
Caustic soda	All iron
Chlorate of lime	Regularly fitted.
Citric acid	Regularly fitted
Copper liquors	Brass fitted
Cyanide	Regularly fitted
Cyanide of potassium	Regularly fitted
Glue (hot)	Brass fitted
Glycerine	Brass fitted
Hydrochloric acid	Lead lined, no zinc
Hyposulphite of soda	Lead lined, no zinc
Lard (hot)	Brass fitted
Magma	Brass fitted, large openings
Mash	Brass fitted, large openings
Milk	All bronze
Milk of lime	All iron
Molasses	Brass fitted, large openings
Naphtha	Regularly fitted
Oil	Brass fitted
Salt water	Brass fitted
Sewage	Brass fitted, large openings
Soap	All iron
Sulphuric acid	All bronze or lead or wood-lined cast-iron, no zinc
Syrup	Brass fitted
Tan liquor	Brass fitted
Tar	All iron, large openings
Vinegar	All bronze
Whiskey	All bronze

By "large openings" is meant that the valve seats must not be obstructed by ribs. Ball valves or clapper valves are suitable for this service.

CHAPTER V

STEAM END DETAILS

103. Slide Valves.—The simplest device for alternately admitting steam to the two ends of the steam cylinder and exhausting it therefrom is the **slide valve** made in the form of the common "*D*" valve. This is shown in Fig. 18. As adapted for a **duplex pump** it differs from that of a **steam engine** in that it is just long enough to cover the ports. It has neither inside nor outside lap except, perhaps, that it is made a trifle longer, say 1/16 inch, so as to cover any irregularity of the edges of the ports.

Any additional lap would increase the possibility of having the valves stop in a position where all ports are covered. This position then constitutes a dead center from which it would be impossible to start the pump. By giving no lap, this danger is practically eliminated, since there is then only one mathematical point of the stroke where the piston must stand to have all ports covered, and **both pistons** would have to stand at their respective dead centers at the same instant.

Table 14.—Dimensions for Slide-valve Motion in Five-ported Cylinder

A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*
1/4	1/4	1/2	3/8	1	15/16	2 3/4	2 13/16	5/8	1/8	.....	13/16
5/16	5/16	5/8	7/16	1 1/4	1 3/16	3 3/8	3 7/16	3/4	3/16	.....	1 1/16
3/8	3/8	3/4	1/2	1 1/2	1 7/16	4	4 1/16	7/8	3/16	.....	1 1/4
7/16	7/16	7/8	9/16	1 3/4	1 11/16	4 5/8	4 11/16	1	1/4	.....	1 1/2
1/2	1/2	1	5/8	2	1 15/16	5 1/4	5 5/16	1 1/8	1/4	1/4	1 3/4
5/8	9/16	1 1/4	11/16	2 3/8	2 5/16	6 1/4	6 5/16	1 3/8	5/16	1/4	2 1/8
3/4	5/8	1 1/2	3/4	2 3/4	2 11/16	7 1/4	7 5/16	1 5/8	3/8	1/4	2 1/2
7/8	11/16	1 3/4	13/16	3 1/8	3 1/4	8 1/4	8 5/16	1 7/8	7/16	1/4	2 7/8
1	13/16	2	15/16	3 5/8	3 9/16	9 1/2	9 7/16	2 1/8	1/2	5/16	3 3/8
1 1/8	7/8	2 1/4	1	4	3 15/16	10 1/2	10 9/16	2 3/8	9/16	5/16	3 3/4
1 1/4	15/16	2 1/2	1 1/16	4 3/8	4 5/16	11 1/2	11 9/16	2 5/8	5/8	5/16	4 1/8
1 3/8	1 1/16	2 3/4	1 3/16	4 7/8	4 13/16	12 3/4	12 13/16	2 7/8	11/16	3/8	4 5/8
1 1/2	1 1/8	3	1 1/4	5 1/4	5 3/16	13 3/4	13 13/16	3 1/8	3/4	3/8	5

Fig. 162 and Table 14 give the principal dimensions for laying out a slide-valve motion for a cylinder of the five-port type. As mentioned before, the author does not recommend this type on account of the large unnecessary clearance, but prefers the type shown in Fig. 38. The above dimensions can be applied to this case also with slight modifications. Not only is the steam waste reduced, but the first cost is also cut down because the steam chest and cover are smaller and the steam cylinder itself is somewhat lighter.

The table is self-explanatory. The dimension L is the travel of the crank pin at contact stroke corresponding with dimension L in Fig. 162. The lost motion which is equal to $2J$ is, of course, the same for either style of valve rod clearance, whether it is made like Fig. 33-a, 33-b, or 33-c.

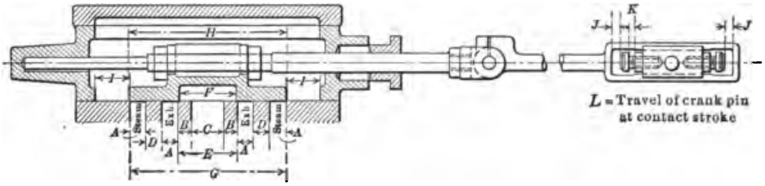


FIG. 162.—Duplex valve motion dimensions.

In compiling Table 14, the following rules were used:

$$L = 3.3A$$

NOTE.—$\frac{L}{2} + \frac{1}{8}$ inch is equal to or less than $A + D$.

This is important in order to prevent the opening of the steam port A by the inner edge of the slide valve to the exhaust which would destroy the effect of the cushion.

$$\text{Lost motion} = 2J = 0.3L = A$$

and,

$$I = \frac{L}{2} + J$$

This is done so as to guard against the slide valve hitting the inside of the steam chest, should the lost motion be put all on one side, inadvertently.

The main drawback of the slide valve is its large unbalanced area which, multiplied by the steam pressure and by the coefficient of friction (about $\frac{1}{6}$), may represent a considerable resistance to be overcome in moving it.

The unbalanced area of the slide valve of a duplex pump consists of the exhaust cavity and one port. This in Fig. 163 would equal $b + a$ multiplied by the length of the port. It is obvious that in any position two ports are open and the third one is balanced, the fourth being unbalanced. As the valve is continually moving it may be supposed to be constantly riding on a film composed of water, steam and oil; therefore the other part of the surface can be assumed to be balanced.

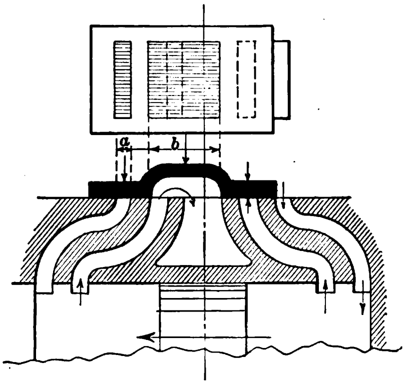


FIG. 163.—Unbalanced area of slide valve.

Various attempts have been made to effect the balancing of the slide valve. Mr. Worthington recognized this important feature and, as early as 1854, fitted an engine built for the *City of Savannah* with a balanced valve. This is reproduced in Fig. 164. The valve is balanced, or rather the superincumbent pressure is reduced, by a piston working in a small cylinder and attached by a suspension bar in the dome of the steam chest. It is worthy of notice that the auxiliary cylinder a is properly vented by a hole v leading to the exhaust

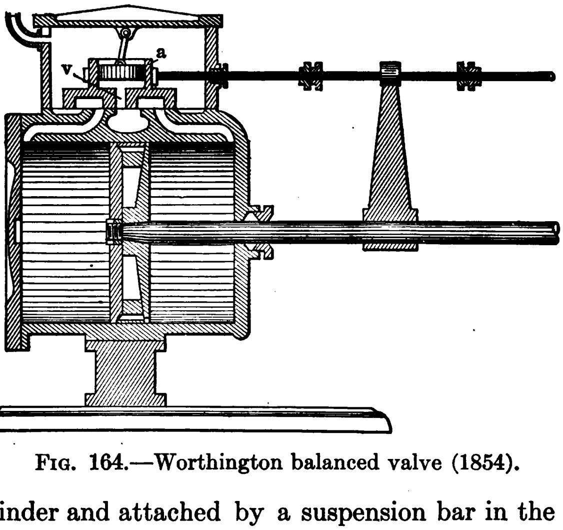


FIG. 164.—Worthington balanced valve (1854).

cavity. The author has several cases on record where expensive balancing devices were put on the slide valve without providing this important feature. Without this vent, of course, the pressure in the auxiliary cylinder quickly builds up and the balancing effect is destroyed, making the balancing device a worthless expenditure.

Another form of balanced slide valve is shown in Fig. 165. Here the valve is enclosed in a box which rests on the steam cylinder face. It is held down by setscrews in the steam chest cover, and is fitted over the valve. This form was used by the Gordon Steam Pump Co. and later by the Laidlaw-Dunn-Gordon Co. of Cincinnati, Ohio. About 75 per cent. of the unbalanced area should be balanced so as not to overdo it and leave a certain unbalanced pressure to keep the valve against its seat.

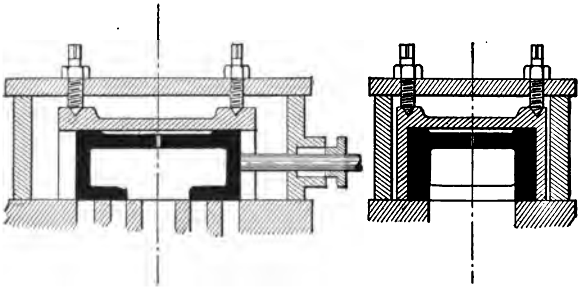


Fig. 165.—Perfectly balanced slide valve.

104. Rotary Valves.—A modification of the slide valve is the rotary valve introduced about fifteen years ago and used chiefly on large engines.

Fig. 166 represents the low-pressure cylinder of a triple-expansion engine. The valve *a* is placed at the bottom of the cylinder so as to make the latter self-draining. The ports *b*, starting at the ends of the cylinder, are extended downward to the under side of the valve, by which arrangement the valve seats from its own weight and is always on the seat whether the steam is on or not. The exhaust port *c* is connected directly with the exhaust pipe which, in this particular case, points toward the inboard end of the cylinder. The reason for this is that in most cases these engines are provided with a condenser located at the water end and the exhaust is thereby more easily connected. It will be noticed that there is only one main port *b* on each end which serves for admission as well as exhaust. For the latter purpose, as was explained in describing the steam cylinder of a duplex pump (article 47), it must terminate a short distance from the head in order to form a cushion.

For admitting the steam behind the piston in starting, a small port *d* is added. A cushion port *e* between the two ports, covered by a cushion valve, furnishes the means for adjusting the cushion.

The self-draining feature, however, makes it impossible to properly lubricate the cylinder through the steam ports, as is usually done, as much of the oil will be washed out by the condensation. While it is true that the oil will mix with the steam and part of it will reach the top of the cylinder, it is clear that the lubricant would be much more effective if introduced on top of the cylinder, as is also indicated in Fig. 166.

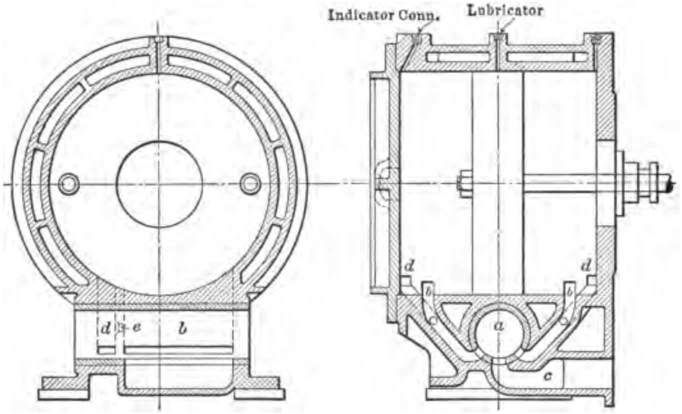


Fig. 166.—Steam cylinder equipped with a single rotary valve.

Fig. 167 shows a longitudinal section through a rotary valve which does not differ from a regular Corliss valve. One end is slotted and into this slot the tee head *b* of the valve spindle is fitted. The cylindrical valve box *c* is covered at each end by a bonnet, one of them, *d*, being provided with a stuffing-box and external guide. A collar *e* is fastened to the spindle and is in contact with the guide for the purpose of taking up the end thrust due to the steam pressure acting on the area of the valve spindle. As this collar is on the outside, it can receive proper attention and lubrication.

Moreover, the cylindrical valve box is bushed, a feature not found in a regular Corliss engine. The reason for this is the number of valve port edges that must be machined true and parallel to each other, as compared with one edge in a Corliss valve box. The bushing is made of cast iron and forced into the valve box.

Although the rotary valve described is only a modification of the slide valve, inasmuch as it is made curved instead of straight, otherwise retaining all the other features, in which is included the absence of any balancing effect, it is nevertheless a valve of much better wearing qualities and seems to be better able to retain the lubricant. It is, therefore, capable of being used with moderately superheated steam for which an ordinary slide valve is unsuitable.

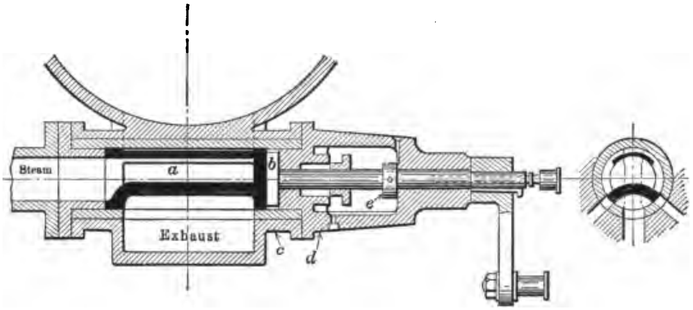


Fig. 167.—Sections of rotary valve and bonnet.

105. Piston Valves.—For working in superheated steam, a valve is required that is perfectly balanced, such as the piston valve shown in Fig. 168. Here, little pistons *a*, fitted with metallic piston rings, are fastened together the proper distance apart so as to match the edges of the ports. The ports run out into an annular space *b* formed around the valve and a bushing *c*, drilled full of holes, is forced in and forms a part of the port, thus preventing the piston rings from springing into the ports and catching on the edges. In this case it is not feasible to use the small auxiliary port, shown in Fig. 38, but the five ports must be used.

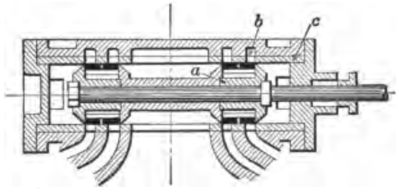


Fig. 168.—Piston valve.

106. Steam Cylinders.—In the following, a few handy formulas will be given for calculating the principal parts of the steam ends:

(a) CYLINDER

$$t = \frac{dp}{4000} + \frac{3}{8}\text{in.}$$

where t = Thickness of wall in inches;
d = Bore in inches;
p = Internal pressure in pounds per square inch.

The steam ends of standard pumps are designed for a pressure of $p = 150$ pounds which substituted in above formula gives:

$$t = 0.0375d + 3/8 \text{ in.}$$

Table 15 gives the wall thicknesses of standard steam cylinders.

TABLE 15.—THICKNESS OF STANDARD STEAM CYLINDERS

Diameter, inches	Thickness, inches	Diameter, inches	Thickness, inches
4-½	7/16	10	13/16
5-¼	½	12	⅞
6	9/16	14	15/16
7	⅝	16	1
8	11/16	18	1-1/16
9	¾	20	1-⅛

(b) COUNTERBORE

No counterbore is required in the steam cylinder of a duplex pump unless some other part, that must be concentric with the steam cylinder, is attached. This is the case in a compound pump where the intermediate head is centered in the counterbore and also when the front head, with its stuffing-box, is attached to the cylinder. If there were no counterbore, the cylinder could not be rebored without destroying this fit. It is the piston ring that wears the shoulder in the cylinder, and although the piston has a fixed position at the extreme end of the stroke in a crank-and-flywheel engine, this is not so in a duplex pump. Therefore, it is not feasible to design the steam cylinder so that the piston ring will always run over the edge of the counterbore. Besides, its length is limited so as not to destroy the cushion and thus most of the time the piston ring will not reach the edge of the counterbore.

(c) Head

Although the steam cylinder head is not in reality a flat plate, on account of its more or less irregular shape it is stronger than such a flat plate and can therefore be considered as such in calculating the thickness. The same rules then apply as were given for calculating the thickness of the water cylinder head; see article 85.

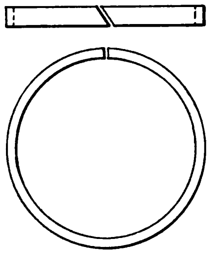


Fig. 169.—Plain snap ring.

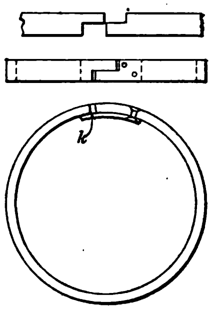
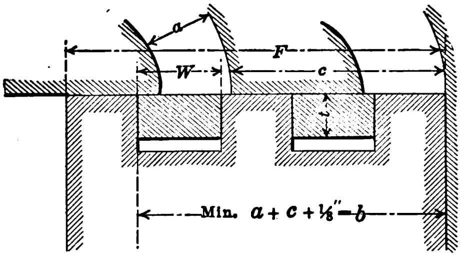


Fig. 170.—Snap ring with keeper.

107. Steam Piston.—The steam pistons are always fitted with self-adjusting metallic packing. This is made in many forms, the cheapest and simplest being the so-called "snap ring" shown in Fig. 169. Here, the ring is turned a little larger than the cylinder bore, is cut open and then forced over the piston body into a groove. It must be cut slanting, as the edges of a straight cut would tend to score the cylinder. Such a ring cannot be expected to be a true fit in the cylinder unless it has been turned after it has been compressed. Even then, this nice fit may be destroyed in springing the ring over the piston body.



$w = a + \frac{1}{8}''$ To Prevent Ring from Catching in Port a

Fig. 171.—Piston ring dimensions.

Snap rings are extensively used on small pistons, but they are poor contrivances, especially if no precautions are taken to stop the leakage through the slot. One way to stop this leakage is to cut the ring "*S*" shaped, as in Fig. 170, thus making the ends overlap. A keeper *k* is then fitted to the inside and fastened to one end, so as to close up the opening.

Sometimes in an endeavor to let every part of the ring bear with the same pressure against the side of the cylinder the ring is made eccentric, and theoretical formulas for such shapes have been devised and can be found in books on the steam engine. It is doubtful, however, whether the wearing qualities of the ring are so much improved thereby as to justify the greater expense. The following rules, with reference to Fig. 171, give satisfactory results for a concentric ring, that is, one of uniform thickness all around:

	Example	
d = Diameter of cylinder bore	12	in.
SNAP RING		
Turn ring to $d + \frac{d}{32}$	12$\frac{3}{8}$	in.
Cut out a piece = $\left(\frac{d}{32} - \frac{1}{16} \text{ in.}\right) \pi$	1	in.
Compress to $d + \frac{1}{16}$ in.	12$\frac{1}{16}$	in.
And turn to d	12	in.
Make thickness of finished ring, $t = 0.0375d$	$\frac{7}{16}$	in.
Width of ring (approx.) $W = \sqrt{\frac{d}{20}}$ 0.775 say	$\frac{7}{8}$	in.
Face of piston (approx.) $F = \sqrt{1.25d}$	4	in.

Rings that are set out by springs, etc., are turned to fit the cylinder bore uncompressed.

To avoid the necessity of forcing the ring over the piston body the piston may be designed with a follower and bull-ring, as shown in Fig. 172. But after having gone to this expense, a better form of ring is generally selected. This may be a ring with a spring and screws, or it may be what is called a three-ring packing, which is shown by Fig. 173. This consists of two outer rings, *a* and *b*, turned to the exact diameter of the cylinder bore and of uniform thickness. A third ring *c*, turned slightly larger than required, is then placed on the inside, and will, by its expansive force, press the two outer rings evenly against the bore. The two outer rings are cut slanting, the joints are staggered, and the inner ring performs the duty of a keeper at the same time. Better and more expensive, are the sectional rings which consist of a number of sections overlapping and set out by small springs.

Two points must be remembered in designing a piston for a duplex pump. Since the exhaust port is some distance from the head, one of the rings will travel over the port and precautions must be taken to prevent it from catching in the port. This may be effected by making the ring at least $\frac{1}{8}$ inch wider than the port, and on very long ports it is well to cast one or two bridges in order to prevent the edges from catching.

The other point is the danger of reopening the exhaust port on the other side of the piston. Fig. 171 shows the piston in its extreme position where it is in contact with the head. If the dimension b, from the face of the piston to the opposite edge of the ring, is made a little longer than the sum of the cushion c plus the port a, or b is as great or greater than $a + c + \frac{1}{8}$ in., there will be no danger of uncovering the port.

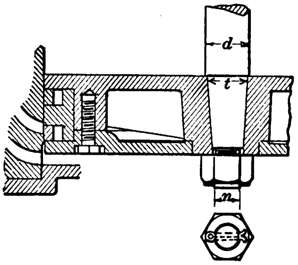


FIG. 172.—Steam piston with bull ring.

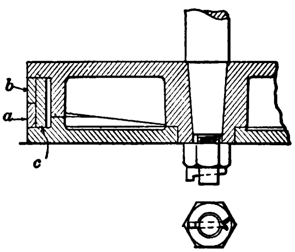


FIG. 173.—Steam piston with three-ring packing.

108. Valve Rods.—The valve rods are made of steel and are usually much stronger than required. Table 16 will serve as a guide to the valve rod sizes, although the actual dimensions may vary greatly from those given:

To the end of the rod is fastened a clevis or fork, provided with a pin, to which the link is attached. This pin is made equal to the diameter of the valve rod, while the crank pin at the other end of the link is made equal to the diameter of the valve rod plus $\frac{1}{8}$ inch. The shank of the link can be made the same diameter as the valve rod.

Since there is considerable lost motion in the connection between crank pin and the slide valve, and it really makes no difference at what point this lost motion is, it seems useless to go to the expense of providing any adjustment in the link ends to take up any lost motion due to wear. Yet, to save the reputation for turning out good workmanship and to prevent all rattling noises due to loose joints, these link ends are sometimes made adjustable. This can be done in various ways, of which the simplest are the best since the wear is so insignificant. Fig. 33c shows a very simple adjustment.

Table 16.—Sizes of Valve Rods

Diameter of steam cylinder, inches	Diameter of valve rod, inches
3	5/16
4½	3/8
5¼	1/16
6	½
8	5/8
10	¾
12	7/8
14	7/8
16	1
18	1
20	1 1/8

109. Rockshafts and Crossheads. On very small pumps the rockshaft, crank and crank pin can be made in one piece, either drop forged or of malleable iron. On larger pumps the parts are made separate, the rockshaft being of steel. The following approximate rule may serve as a guide to the dimensions:

$$r = 0.5\sqrt{d}$$

where r = Diameter of rockshaft;
d = Diameter of steam cylinder.

Each end of the rockshaft is fitted with an arm, a short one to drive the steam valve and a long one for transmitting the motion of the piston rod to the rockshaft. As the end of the arm swings in a circle a crosshead must be interposed. In small pumps this takes the form of a simple spool (see Fig. 174), which is machined out of a round piece of steel. Fig. 175 shows a spool with round flanges, which has the advantage that the piston rod is free to turn. If the flanges are made octagonal or if a hexagon is added, the spool can be held with a wrench when tightening the piston nut. The arm may be forked so as to present two wearing surfaces to the spool.

A more elaborate device will be noted by referring back to Fig.

32. Here a block is made to slide in the groove and carries a pin to which the long arm is pivoted. Fig. 176 is also a good arrangement. The arm is turned round for a short distance from its end and moves through the head of a pin which is swiveled in the crosshead.

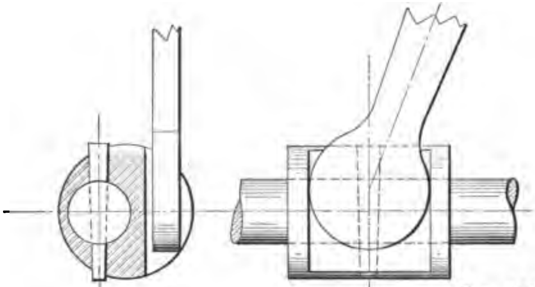


Fig. 174

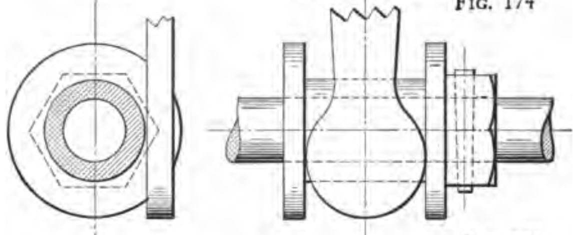


Fig. 175

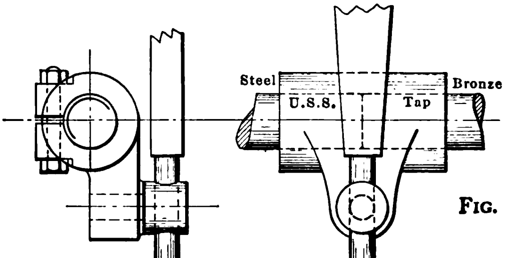


Fig. 176

Figs. 174 to 176.—Crossheads.

None of these devices is complicated by means for adjustment to take up the wear, nor is any such adjustment necessary. In the first place, the wear will be insignificant on account of the slow speed and should any looseness develop, it would have no detrimental effect on the action of the pump, except that it might cause some rattling. On large and important machines the connection is made in the well-known manner with pins and a link. These and other devices can be studied in the numerous illustrations.

On large brass-fitted pumps it is sometimes desirable to split the piston rod at the crosshead and to make only the plunger rod of tobin bronze. This is not done so much to save the extra price of the brass as it is due to a steel rod being far superior for use in steam. Brass is apt to soften in the high temperature and does not wear so well as steel. The ends of the rod are then threaded with coarse threads (U. S. standard) and are locked by splitting the crosshead and clamping it over the rod ends, as shown in Fig. 176.

110. Drain Valves.—Drain valves, which in a steam engine form an important part of the equipment, are not of the same importance in a direct-acting pump. If the piston encounters an extra resistance in having to push some accumulated water out of the cylinder it will merely be retarded in its action or stop completely without wrecking the cylinder. All cylinders, however, are fitted with drain valves for the purpose of draining the cylinder in stopping and starting.

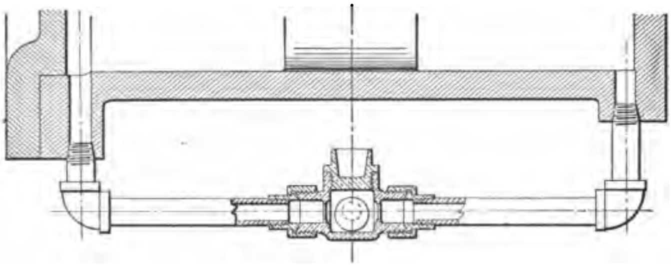


FIG. 177.—Automatic cylinder drain.

If the pump is running very slow or if its action is intermittent it is sometimes advisable to remove the water of condensation automatically. For this purpose the two ends of the cylinder are connected by a pipe containing a double-ended ball drain valve, as shown in Fig. 177. The valve chamber is connected to a steam trap and the ball valve, which is made of hard rubber, will prevent communication between the two ends while the end that is under pressure will drain into the trap.

111. Regulating Devices.—While a regulator is indispensable. in a steam engine, this cannot be said of the direct-acting pump, Here the machine is itself a regulating element, for if the water consumption diminishes, the water pressure rises and the pump slows down. As a rule, the head against which the pump is working is fairly constant, so that this range of regulation is all that is required. Some services, however, require a closer regulation and some suitable regulating device must then be provided.

According to the service for which the pump is being used several types of regulators have been developed, the most important one being the *pressure regulator*. This is used in direct service; that is, an installation where the pump delivers directly into the main pipe. A balanced valve is placed in the steam pipe and is controlled by a spring-loaded piston, plunger or diaphragm in a chamber which is in communication with the main discharge pipe. If the water pressure rises this increased pressure acts on the piston and closes the balanced valve, which slows down the pump until the normal water pressure is restored, when the balanced valve returns to its former position. This type is used in waterworks employing the direct system, in pumps supplying hydraulic presses and in fire pumps.

An improved form is one in which the cylinder operating the balanced valve is provided with ports and a pilot valve controlled by the spring-loaded piston. This admits of a wider range of regulation and it has the advantage that the balanced valve does not return to its original position after the normal pressure is restored, but will find the position required to maintain the normal water pressure under the changed conditions.

Another type of regulator is the *speed governor*. This is used where the head is constant, as in reservoir service, and where a safeguard against the pump running away is desired. This consists of a balanced valve controlled by a little oil pump which is driven by any reciprocating part of the valve motion. If the speed increases the oil pump delivers more oil than can flow back through an adjustable needle valve and thus closes the balanced valve. This device is embodied in the design shown by Fig. 103.

In elevator service the demand for water is variable and often intermittent; hence special regulating devices have been developed for this service which will be taken up more in detail under "Elevator Pumps."

CHAPTER VI

SERVICE CONDITIONS VS. VARIOUS TYPES

112. Boiler Feed Pumps.—The unit *boiler horsepower* was established by the judges of the Centennial Exposition, in 1876, to be the evaporation of 30 pounds of water at 100 deg. F. into steam at 70 pounds gage pressure, which is equivalent to 34.5 pounds evaporated from and at 212 deg. A boiler, however, that is rated according to this standard is generally provided with sufficient heating surface to evaporate much more water than is indicated by this rule, and since it is the duty of the feed pump to furnish at least as much water as is evaporated in the boiler, and sometimes more, if the water level is low and must be raised quickly, this greater capacity should be taken into consideration in calculating a boiler feed pump.

Practice among boiler manufacturers is divided as to the heating surface allowed per nominal horsepower. Most water-tube boilers are now rated at 10 square feet per horsepower, while makers of fire-tube boilers, as a rule, hold to the older rating of 12 square feet per horsepower. As there is little decrease of economy if the boiler is forced to evaporate 4 pounds[1] of water per hour per square foot of heating surface, this would make the evaporation equal to 48 pounds per hour per nominal horsepower.

The Committee of the American Society of Mechanical Engineers on "Trials of Steam Boilers" reported that a boiler should be capable of developing its rated horsepower with easy firing, moderate draft, and ordinary fuel, and furthermore, that it should be capable of delivering at least one-third more than its rated power to meet emergencies.

These considerations have led to the adoption of the rate of 45 pounds per hour per horsepower, as the quantity for which the boiler feed pump should be calculated. The quantity thus obtained should be delivered to the boiler at a moderate speed so that in case of low water in the boiler, the pump could be speeded up and the deficiency made up promptly. Therefore, it is good practice to reduce the speed of the feed pump to one-half that of a pump used on regular service.

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[1] This is exceeded in many large central stations in everyday operation.

Table 17 gives the standard sizes of duplex direct-acting boiler feed pumps with their capacities in gallons per minute, pounds per hour, and boiler horsepower, at the 45-pound rating.

TABLE 17.—CAPACITIES OF BOILER FEED PUMPS

Size, inches	Gallons per revolution	Revolutions per min.	Piston speed, ft. per min.	Gallons per min.	Pounds per hour	Horse-power of boiler supplied at 45-lb. rate
3 × 2 × 3	0.155	40	20	6	3,000	67
3½ × 2¼ × 4	0.265	37.5	25	9.6	4,800	106
4½ × 2¾ × 4	0.395	37.5	25	14.4	7,200	160
5¼ × 3½ × 5	0.805	36	30	28	14,000	310
6 × 4 × 6	1.265	32.5	32.5	39.6	19,800	440
7½ × 4½ × 6	1.6	32.5	32.5	50.6	25,300	560
7½ × 4½ ×10	2.66	24	40	60	30,000	670
8 × 5 ×10	3.25	24	40	76	38,000	850
9 × 5¼ ×10	3.6	24	40	84	42,000	930
10 × 6 ×10	4.75	24	40	110	55,000	1,220
12 × 7 ×12	7.75	22.5	45	170	85,000	1,880
14 × 8½ ×12	11.5	22.5	45	250	125,000	2,780
16 ×10¼ ×12	16.7	22.5	45	360	180,000	4,000
18 ×12 ×12	23.0	22.5	45	500	250,000	5,550
20 ×14 ×12	31.5	22.5	45	690	345,000	7,650

Any type of pump can be used for this purpose—plunger and ring, piston, outside center-packed or a pressure pump. They may also be made compound, where economy is an object, or vertical where the floor space is limited. Centrifugal pumps also are sometimes employed; but as these do not come within the scope of the present work they will not be considered here.

As a rule high economy is not any advantage, because the steam used is utilized to heat the feed water. Thus the latent heat of the exhaust steam is regained and the heat actually expended in pumping the water into the boiler becomes a negligible quantity.

It is desirable to remove the condensate in engine jackets and many heating systems as fast as it is formed, and pump it back into the boiler. This may be accomplished automatically by the "automatic feed pump and receiver" illustrated in Fig. 178. The condensate flows into a tank which is provided with a float operating a balanced valve in the steam pipe of the pump. If the water level rises in the tank the float opens the valve and starts the pump which delivers the water into the boiler until it has reached a level at which the valve is closed.

The economy in this outfit is greater the higher the steam pressure in the system, because the water is pumped back into the boiler at the temperature of the steam. If a trap is used the temperature is naturally reduced to 212 deg. in blowing into the hot well and the difference is lost.

The size of the pump is calculated on the basis of ½ pound of steam condensed per hour per square foot of radiating surface at a slow speed equal to one-half of that for regular service.

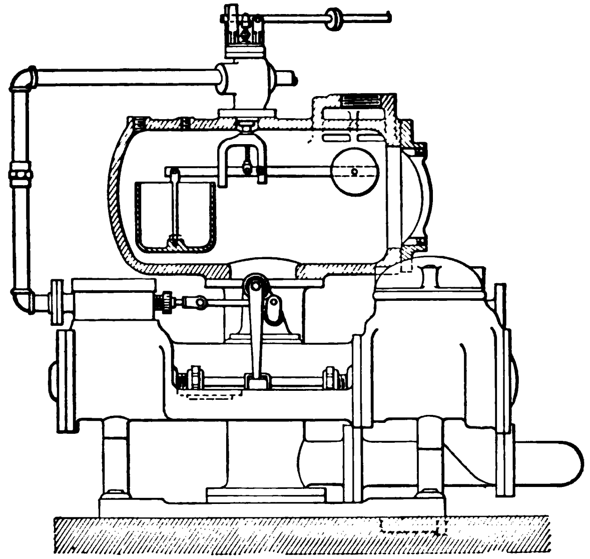

FIG. 178.—Automatic feed pump and receiver.

113. Condenser Pumps.—After the steam has done its work in the cylinder, it enters the condenser where it is liquefied by coming in contact with cold water, either directly as in the jet condenser, or indirectly through water-cooled surfaces as in the surface condenser.

Water under atmospheric conditions contains about 2 per cent. air. This is pumped into the boiler with the feed water and is there compressed into an insignificant volume. In passing through the engine and expanding, the quantity is increased by leakage into the stuffing-boxes of the low-pressure cylinder, joints, etc., so that by the time the steam reaches the condenser it may contain 5 per cent. or more of air (expressed in percentage of the condensate by volume).

With a vacuum of 27 inches, or an absolute pressure of 3 inches, this air will expand to ten times its volume under atmospheric condition. Therefore, considering 1 cubic foot of feed water, the space formerly occupied by 0.95 cubic feet of water and 0.05 cubic foot of air will now occupy 0.95 cubic foot of water and 0.5 cubic foot of air, making a total of 1.45 cubic feet. This shows that the air now occupies 35 per cent. of the total space, and the condensate will enter the pump in this condition. The pump must compress this air to atmospheric pressure in order to discharge it to the atmosphere. The pump is, therefore, more of an air compressor than a water pump and a duplex pump will not do at all for this purpose. There being no resistance at the beginning of the stroke, except friction (see Fig. 179), one piston of a duplex pump would move rapidly forward and throw the valve of the opposite side, before the piston of the opposite side, which has been laboring against a high resistance at the end of

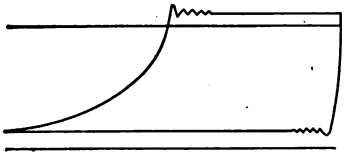

Fig. 179.—Indicator diagram of wet vacuum pump.

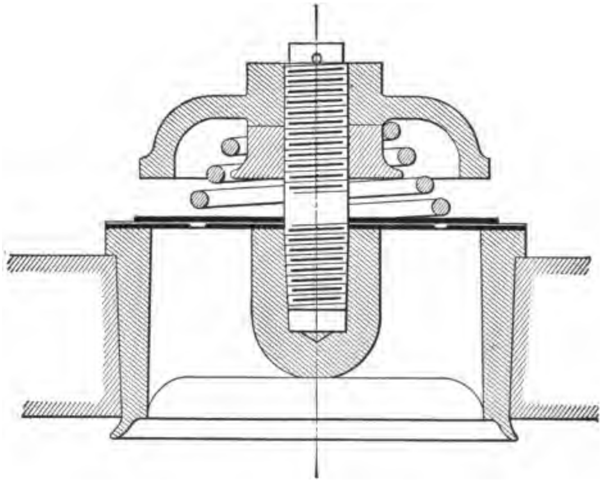

Fig. 180.—Kinghorn valve.

the stroke, would have time to finish its stroke. The result will be a short stroke and a very low efficiency.

For this purpose, therefore, single pumps are preferred. Here the piston cannot possibly reverse until it has completed its stroke, and, with moderate piston speeds, the result is a quietly working, effective pump.

The pump ends, even in large sizes, are of the submerged piston pattern because there is no clearance, which in an air pump would be highly detrimental. The water contained in the pump body and the ports forms a water piston traveling back and forth with the pump piston and this water, being subjected to a vacuum most of the time, is void of air. Its upper level, in receding from the discharge valve deck, leaves a nearly perfect vacuum behind. This, of course, is soon impaired by the mixture of air and water coming in through the suction valve. During the reverse stroke this air-saturated water is then discharged before the air has had an opportunity to penetrate the water that has receded into the ports. The reversal of this heavy mass of water makes it impossible for the pump to run fast. The proper speeds for running wet-vacuum pumps are given under "efficiencies."

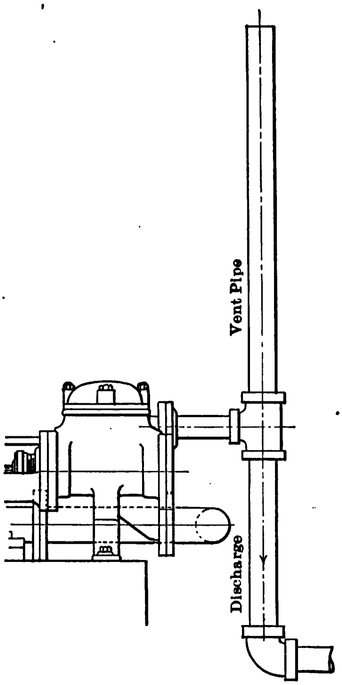

FIG. 181.—Air vent pipe.

The pump ends are fitted with soft rubber valves, or "Kinghorn" valves; see Fig. 180. The latter consists of a number of manganese-bronze disks of different diameters, the lower ones having a few holes to prevent their adhering together, and to effect a successive opening.

No air chamber is required, but where the discharge turns down, a vent pipe (Fig. 181) should be placed on the elbow so as to release all the air. It may then be considered that the whole atmosphere forms an air chamber.

114. Pumps for Jet Condensers.—For ordinary service, such as a 26-inch vacuum and a cooling-water temperature of 70 deg., the displacement D of the wet vacuum pump with a jet condenser is made equal to

$$D = W + A + S$$

where W = Amount of injection water;

S = Amount of condensate;

$A = \frac{2}{p_a}W$ = Amount of air and vapors arising from the water, all in cubic feet;

P_a = Absolute pressure in inches of mercury.

In calculating the amount of injection water, let H be the total heat in 1 pound of the steam from zero degrees as it enters the condenser. This varies little under ordinary conditions, as is shown by the following values:

Terminal pressure	Total heat, B.t.u.	
	From 32 deg. F.	From 0 deg. F.
6 lb.	1132	1164
10 lb.	1141	1173
20 lb.	1151	1183
30 lb.	1158	1190

By assuming a total heat of 1200 B.t.u. the error will be on the safe side.

Let t_c be the temperature of the injection water as it enters, say 70 deg., and t_v the theoretical temperature of the steam at the desired vacuum; for a vacuum of 26 inches, this would be t_v = 125 deg. F. A perfect condition would prevail if the mixture of water and steam could be discharged at this temperature. Such conditions, however, cannot be expected anywhere; therefore a little more water than the theoretical amount must be injected, which will cool the mixture to a temperature below the theoretical temperature, say to 120 deg. This would then be the temperature of the air-pump discharge, or t_d = 120 deg. The difference between these two temperatures will always be around 5 deg. Hence

$$t_d = t_v - 5$$

Each pound of injection water will then be heated from 70 to 120 deg., and will absorb 120 − 70 = 50 B.t.u.; and the total heat absorbed by the water will be

$$H_w = W(t_d - t_c)$$

The heat given up will be

$$H_s = S(H - t_d)$$

In this case for 1 pound of steam

$$H_s = 1200 - 120 = 1080 \text{ B.t.u.}$$

Equating the two expressions, since the heat absorbed must equal that given up,

$$W(t_d - t_c) = S\,(H - t_d)$$

whence,

$$W = \frac{S(H - t_d)}{t_d - t_c}$$

A more convenient form is:

$$w = \frac{W}{S} = \frac{H - t_d}{t_d - t_c}$$

where w represents the factor by which S must be multiplied in order to obtain the amount of injection water, W.

In the example,

$$w = \frac{W}{S} = \frac{1200 - 120}{120 - 70} = \frac{1080}{50} = 21.6$$

That is, for every pound of steam condensed, 21.6 pounds of water must be injected. The amount of air and vapor arising from 21.6 pounds of injection water for a 26-inch vacuum ($p_a = 4$), will then be per pound of steam:

$$a = \frac{2}{4}w = 10.8$$

and the displacement of the wet vacuum pump will be:

$$\begin{aligned}(w + a + 1)S &= (21.6 + 10.8 + 1)S \\ &= 33.4S \text{ lb.}\end{aligned}$$

or, say thirty-four times the amount of steam.

Example.—The amount of steam to be condensed in a jet condenser per hour is 7000 pounds, the vacuum 26 inches, and the injection-water temperature 70 deg. F. Find the size of air pump.

Solution. $W = 34 \times 7000 = 238{,}000$ lb. per hr.

$$G = \frac{238000}{8.33 \times 60} = 476 \text{ gal. per min.}$$

Selecting a single pump running at a moderate speed of, say 72 feet per minute (see Table 1, page 16) it will be found that a 14-inch piston is required. This can also be found by applying the formula.

$$d^2P = 24.5G$$

$$d = \sqrt{\frac{24.5G}{P}} = \sqrt{\frac{24.5 \times 476}{72}}$$

$$d = 12.4, \text{ say } 14 \text{ in.}$$

The commercial size of the air pump will then be 10 × 14 × 12, which means that the pump will have a single steam cylinder of 10 inches diameter and 12 inches stroke, and a pump piston, 14 inches diameter and 12 inches stroke.

115. Pumps for Surface Condensers.—A surface condenser requires two pumps, one for supplying the circulating water and the other for removing the air and hot water (condensate). In fact, where a very high vacuum is desired, as in steam turbine work, the air is removed separately by a dry-vacuum pump. The condensate, or hot water, collects in a hot-well from which it is removed by an ordinary water pump.

The circulating and the wet-vacuum pumps are frequently combined into one unit, called a *combined air and circulating pump*, the steam cylinder being placed in the middle between the two pump cylinders, as shown in Fig. 182.

In calculating the size of the circulating pump for a surface condenser a greater allowance in temperature must be made than in the case of the jet condenser, because here the steam and water do not come in immediate contact but are separated by the walls of the tubes. A difference of 15 deg. is a fair allowance. As before, let

H = Total heat (above 0 deg.) of steam entering condenser;
t_v = Theoretical temperature at vacuum;
t_c = Temperature of cooling water at inlet;
t_d = Temperature of cooling water at outlet;
W = Weight of condensing water;
S = Weight of condensed steam.

Then

$$t_d = t_v - 15 \text{ deg.}$$

$$w = \frac{W}{S} = \frac{H - t_v}{t_d - t_c}$$

where w now represents the factor by which S must be multiplied in order to obtain the amount of cooling water.

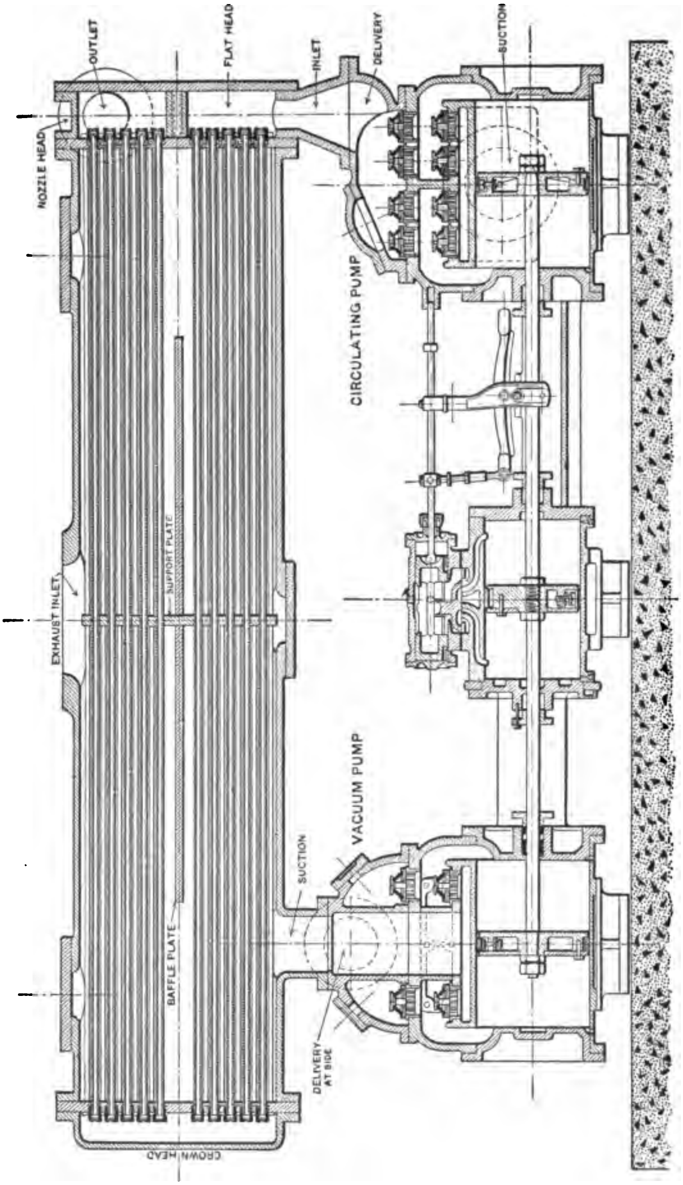

FIG. 182.—Surface condenser mounted over single air and circulating pump.

Example.—Assume 7000 pounds of steam to be condensed per hour at a vacuum of 26 inches, which has a theoretical temperature of t_v = 125 deg., and with a cooling-water temperature of t_c = 70 deg. Find the size of the air and circulating pumps.

$$t_d = t_v - 15 = 125 - 15 = 110 \text{ deg.}$$

Then,

$$W = \frac{H - t_v}{t_d - t_c} = \frac{1200 - 125}{110 - 70} = 27 \text{ (approximately)}$$

which means that 27 pounds of water is required for every pound of steam. Hence $W = 27 \times 7000 = 189{,}000$ lb. per hr. To convert this into gallons per minute divide by 500, because 1 gal. per min. = 8.33 × 60 = 500 lb. per hr.

Then

$$G = \frac{189{,}000}{500} = 378 \text{ gal. per min.}$$

If a pump of 12-inch stroke is selected, from Table 1, page 16, the proper speed of 72 feet per minute will be found and the size can be calculated by the formula:

$$d^2P = 24.5G$$

from which

$$d = \sqrt{\frac{24.5 \times 378}{72}} = 11.3$$

or say,

$$d = 12 \text{ in. diameter.}$$

The wet-vacuum pump is not so easily calculated because it operates as an air compressor and the amount of water it has to handle is very little, just enough to seal the valves. Since a direct-acting pump is a very imperfect air compressor, a large allowance must be made over the theoretical size. Therefore, it is good practice to make the capacity of the air pump for a surface condenser equal to from twenty to thirty times the amount of steam to be condensed, expressed in pounds per unit of time. Which multiple is to be used will depend upon the service, the vacuum required and other conditions. The following formula may serve as a guide. Displacement of wet vacuum pump

$$= 1.5S\left(\frac{54}{P_a} + 1\right)$$

where $\frac{54}{P_a}$ represents the amount of vapor arising from the steam.

In the example this would mean a factor: $1.5\left(\frac{54}{4} + 1\right) = 21.5$ and a capacity of

$$7000 \times 21.5 = 150{,}500 \text{ lb. per hr.}$$

or

$$G = \frac{150{,}500}{500} = 301 \text{ gal. per min.}$$

At 72 feet piston speed this would require a 10-inch cylinder, for which we could then select a 10-inch pump, making the size of the air and circulating pump

$$10 \times 10 \times 12 \times 12 \text{ in.}$$

It means that the pump will have one 10-inch steam cylinder, one 10-inch air cylinder, one 12-inch circulating water cylinder, and a common stroke of 12 inches.

A 10-inch steam cylinder would be ample to drive the pump under ordinary conditions. In this connection, it would be fair to assume a pressure of 15 to 20 pounds in each pump cylinder, a steam pressure of 100 pounds and a mechanical efficiency of 50 per cent. Should the steam pressure be low, or the friction through the condenser and pipes very high, the size of the steam cylinder must be calculated carefully.

116. Waterworks Condensers.—As the subject of surface condensers is intimately connected with the manufacture of direct-acting pumps a few words on the problems that have to be solved in this respect may be permissible.

In waterworks service where an abundance of cool water is always available, the surface condenser is frequently built into the suction or discharge pipe. The condenser then differs from the regular type in that the cooling water flows around the tubes and the exhaust steam is condensed inside the tubes, the object being to provide ample passage for the water. In a duplex pump with separate cylinders the cross-suction pipe is utilized for this purpose. It is made large in diameter and the open ends are fitted with bronze or cast-iron tube heads and crown heads to which the main exhaust pipe is connected on one side and the air pump suction on the other. These condensers are always built with one pass. Fig. 183 shows a surface condenser located in the cross-suction pipe of a triple-expansion pump.

In designing a waterworks condenser, care must be taken to keep the water openings clear and allow ample waterway around the tubes, a point that is often neglected by inexperienced designers. In this case it may happen that the water cylinders do not fill, which is indicated by a short jump of the plungers at beginning of stroke.

The recognized surface condenser construction calls for tube heads of composition and tubes of seamless drawn brass, untinned. These tubes pass through stuffing-boxes in both tube heads, which have threaded brass ferrules, each with a lip on the inside as shown in Fig. 184. This lip prevents the tubes from creeping, which is a result of the constant expansion and contraction of the tube and shell.

If the discharge pressure is moderate the condenser is preferably located in the cross-discharge pipe. Although it will be more costly here, on account of heavier construction, there will be the advantage of having no obstruction in the suction pipe, which is especially to be avoided if the suction lift is high. A further advantage is that the condenser is placed some distance above the floor line, which gives a good head on the wet vacuum pump. For the case of a suction condenser, this pump must necessarily be placed in a pit.

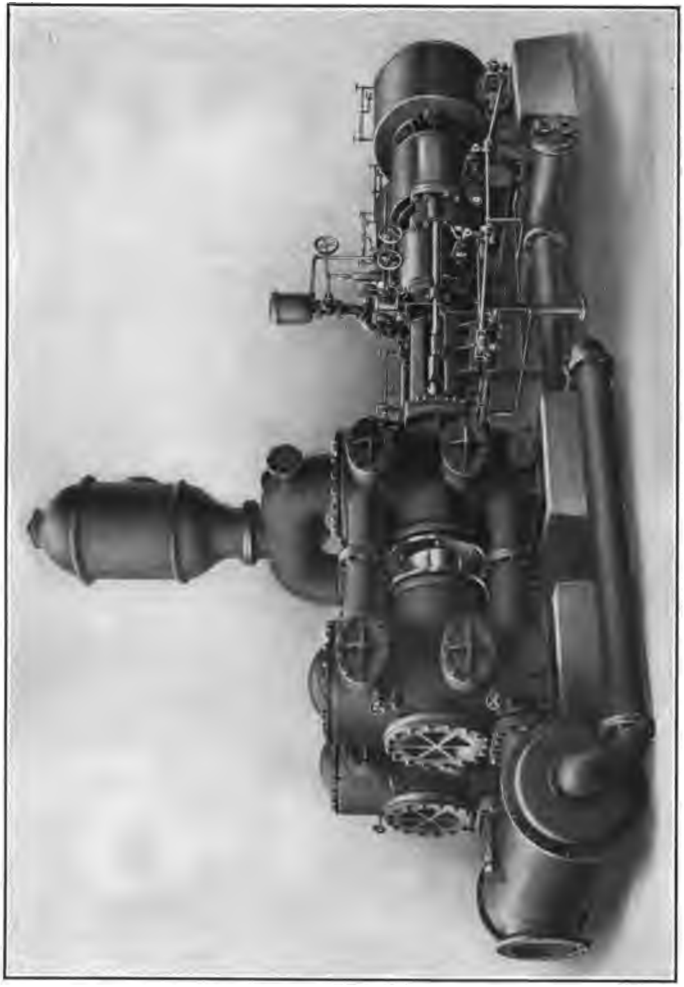

Fig. 183.—Waterworks pumping engine with surface condenser in suction.

Since the amount of cooling water is fixed in the case of a waterworks surface condenser, the procedure in calculating the surface required is somewhat different from that employed for an **ordinary condenser,** where we allow a heat transference of 300 B.t.u. per square foot per hour per degree mean difference of temperature (D).[1] The latter is the average difference between the temperatures at both sides of the cooling surface, and for present purposes may be taken roughly as:

$$D = t_v - \frac{(t_d + t_c)}{2}$$

t_v, t_d and t_c representing the same items as before.

This expression multiplied by 300 will give the heat in B.t.u. transferred per hour by 1 square foot, which must equal the heat given up by the steam, namely, $s(H - t_v)$, where s is the weight of steam condensed by 1 square foot per hour and H and t_v are as before. Therefore,

$$s\,(H - t_v) = 300\left[t_v - \left(\frac{t_d + t_c}{2}\right)\right]$$

and

$$s = \frac{300\left[t_v - \left(\frac{t_d + t_c}{2}\right)\right]}{H - t_v}$$

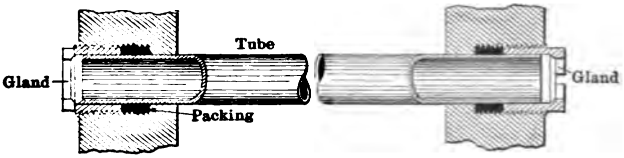

FIG. 184.—Condenser tube and packing.

For a vacuum of 26 inches and a cooling water temperature of 70 deg., with $t_v = 125$ deg. and $t_d = 125 - 15 = 110$ deg.,

$$s = \frac{300\ (125 - 90)}{1075} = 9.75$$

or, practically 10 pounds of steam condensed per square foot of

---

[1] Strictly speaking, the mean temperature difference between the water and the steam is,

$$D = \frac{t_d - t_c}{\log_e \dfrac{t_v - t_c}{t_v - t_d}}$$

surface per hour. Therefore, 7000 pounds of steam per hour would require a regular surface condenser of

$$\frac{7000}{10} = 700 \text{ sq. ft. of surface.}$$

In the foregoing calculation the rise in temperature ($t_d - t_c$), of the cooling water was given. In the case of a **waterworks surface condenser,** however, the amount of cooling water is given and the rise in temperature must be calculated. The following relation must then exist:

$$\frac{W}{S} = \frac{H - t_v}{t_d - t_c}$$

Where W and S represent respectively the weights of water and condensed steam. Hence the temperature rise of the water is:

$$t_d - t_c = \frac{S}{W}(H - t_v)$$

In general this rise will be about 4 deg. F.; therefore,

$$s = \frac{T[t_v - (t_c + 2°)]}{H - t_v}$$

where T represents the heat transferred per square foot per hour per degree of mean difference. This was assumed for a regular condenser to be 300 B.t.u., but in a waterworks condenser much of the water is flowing by the tubes without having any cooling effect. Therefore, in this case T will be taken as 250 B.t.u. per square foot per hour per degree of mean temperature difference, which makes the formula:

$$s = \frac{250[t_v - (t_c + 2°)]}{H - t_v}$$

For $t_c = 70$ deg. and a 26-inch vacuum we have

$$s = \frac{250 \times 59}{1075}$$

$s = 12.4$ pounds of steam condensed per hour per square foot of surface.

A rise of 4 deg. was assumed. This, however, will depend upon the economy of the pump and the water pressure. If the latter is high the steam consumption is high; the same effect means low economy. High water pressure and low economy, therefore, demand a larger surface condenser than a high economy and low water pressure.

The economy of a steam pump, as will be explained under "Duty," is generally expressed in foot-pounds of work done per 1000 pounds of steam used. The cooling effect of a condenser can then be expressed in terms of duty and water pressure, and Fig. 185 shows a diagram that may be found useful in this respect. The abscissæ represent duties from zero to 200 million foot-pounds per 1000 pounds of steam, and the ordinates, water pressures from zero to 300 pounds per square inch. The diagonal lines then give the amount of steam condensed in pounds per hour per square foot of cooling surface.

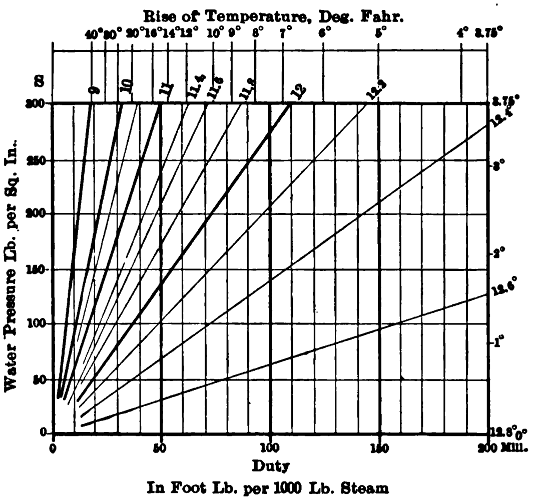

FIG. 185.—Surface condenser rating in pounds per hour per square foot on domestic service.

Example.—Assume a duty of 80 million foot-pounds and a water pressure of 110 pounds per square inch. Find the amount of steam condensed per hour per square foot.

Locate 80 million on the horizontal reference line and 110 pounds on the vertical; then follow the vertical and horizontal guide lines and where they intersect read 12.4 pounds of steam condensed per hour per square foot. The horizontal line on top and the last vertical line give the rise of temperature in the water, which in this case would be 3.5 deg. F.

This diagram is based on a vacuum of 26 inches, a water temperature of 70 deg. F. and a heat transference of 250 B.t.u. per hour per square foot per degree of mean temperature difference.

It often happens that a compound or triple-expansion engine is fitted with live steam connections to admit live steam to the larger cylinder in case of fire. The pump is then running compound on domestic service and high pressure on fire service, or triple-expansion in one case and compound in the other. When running against fire pressure the horsepower is then greater and also the steam consumption per horsepower-hour, which results in the need for a very large surface condenser. In this exceptional case, however, a high vacuum is not absolutely necessary and the diagram in Fig. 186, based on a 22-inch vacuum, can be employed. If then the duty, running on fire service, is 50 million foot-pounds and the fire-service pressure is 150 pounds, 1 square foot of surface will be found to condense 18.6 pounds of steam per hour. The steam consumption in pounds per hour divided by this figure will give the size of the surface condenser in square feet. In using this diagram, the duty, when running on fire service, must be known. This can be calculated by the rules given in Section 129. First calculate the new steam pressure in the low-pressure cylinder

(1) Compound: $$p_1 = \frac{\text{m.e.p.} + bR - p}{R - 1}$$

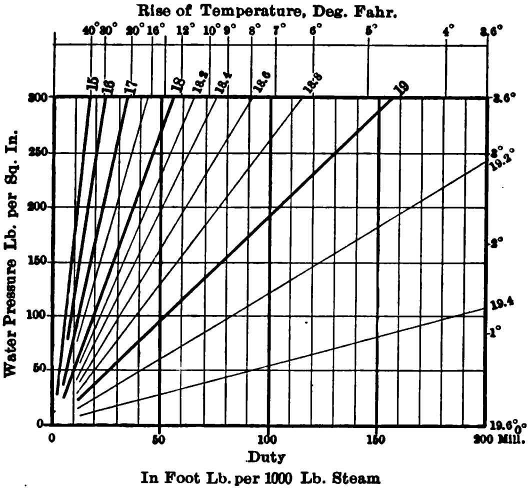

FIG. 186.—Surface condenser rating in pounds per hour per square foot on fire service (22-inch vacuum).

where, m.e.p. = mean effective pressure referred to the high pressure cylinder area.

b = 8 pounds for a 22-inch vacuum.
R = Ratio of cylinders.

Then,

$$\text{Card Duty} = 144000 \times \frac{\text{m.e.p.}}{R} V$$

where V = Specific volume at p_1. Make allowances as given in Tables 18 to 24 (pages 232 to 238).

(2) Triple-expansion: $$p_1 = \frac{\text{m.e.p.} + bR_1 - p}{2(\sqrt{R_1} - 1)}$$

where, m.e.p. = mean effective pressure referred to the high-pressure cylinder area, and,

b = 7 pounds for a 22-inch vacuum.
R_1 = Ratio of low-pressure to high-pressure cylinder.

Then,

$$\text{Card Duty} = 144000 \frac{\text{m.e.p.}}{\sqrt{R_1}} V$$

where V = Specific volume at p_1.

117. Attached Air Pumps.—If it is not desired to utilize the exhaust of the independent air pump for heating the feed water, this pump may be attached to some reciprocating part of the larger pump and thus the economy, as a whole, is improved. There are various ways of attaching the air pump, one of the best being to make it duplex and drive each side from one of the two rockshafts of the main pump. These, of course, must be made extra strong to take care of the added load. Fig. 187 shows such an arrangement in which a vertical air pump is driven by means of the cranks attached to the rockshafts. The capacity of the attached air pump is calculated on the basis of a displacement equal to thirty times the amount of steam to be condensed.

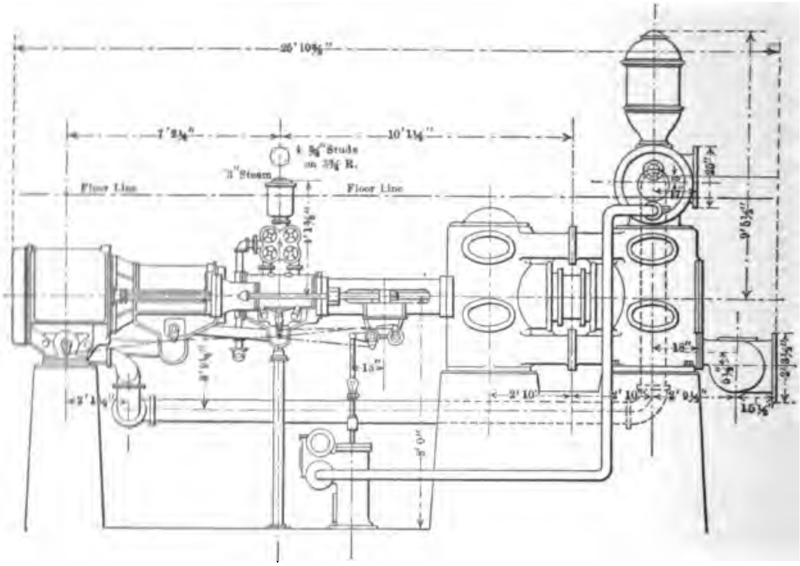

Fig. 187.—Triple-expansion pump with surface condenser in discharge and attached air pump.

118. Feed-water Heaters.—To utilize part of the heat in the exhaust steam before it is removed by the condenser, a feed-water heater is often inserted at the point where the exhaust leaves the low-pressure cylinder. It is generally built into the low-pressure cross-exhaust pipe as in Fig. 183. The boiler-feed pump discharge is connected to the lower opening and the upper opening is piped into either an open or a closed auxiliary exhaust feed-water heater which is heated by the exhaust steam coming from the auxiliary pumps, such as boiler feed, air pump, etc.

The main exhaust feed-water heater is calculated on a basis of 80 pounds of steam per hour per square foot, and the closed auxiliary exhaust heater may be made one-half the size of the main heater.

The heater is built like a surface condenser only stronger, and the tubes are made of wrought iron, the steam flowing around the tubes and the feed water through them.

119. Elevator Pumps.—These pumps are used for supplying water to hydraulic elevators. They are usually located in basements and confined to a small floor space; hence must be of compact design. They are invariably run noncondensing. As the amount of water used by the elevators varies greatly, these pumps must be equipped with a governing device that changes the speed in relation to the amount of water consumed.

Direct-acting pumps are admirably adapted for this service. They merely require a pressure regulator that keeps the pressure in the tank uniform. Any sudden demand will lower the pressure in the tank sufficiently to start the pump, and any stoppage of the elevators will increase the pressure in the tank sufficiently to stop the pump.

As previously explained, a direct-acting pump is ready to start from any position as soon as steam is admitted to the cylinder. In the case of a triple-expansion engine, however, the full steam pressure acting on the small high-pressure piston may not be sufficient to start the pump against a full load. The water end must, therefore, be relieved of its pressure as soon as the pump stops. This may be accomplished by installing a check valve in the discharge pipe, which, by closing, opens a small by-pass valve in the force chamber and releases enough water to relieve the chamber of all pressure. It is essential that the air chamber be placed outside of this check valve, for otherwise, the entire air would be lost through the bypass every time the pump stopped; besides, it would take a long time for the force chamber to relieve itself. With the air chamber shut off from the pump body, the removal of only a small amount of water is sufficient to remove the water pressure.

When the elevators start again, the pressure in the discharge tank drops, causing the pressure regulator to admit steam to the pump, which begins to move slowly, at first, discharging all the water through the little bypass valve. As the pump gains in speed, this bypass valve cannot pass all the water pumped and the check valve opens, thereby closing the bypass.

Where there are many elevators, their combined action results in practically a uniform water consumption and the pump is then kept running at a uniform speed. In this case, such contrivances can be dispensed with and the pressure regulator can be jacked up so that it will not stop the pump entirely, but will keep it running at a slow speed. Should the elevators stop for a considerable time, the surplus water is discharged back into the tank through a relief valve.

Elevator pumps use the same water over and over again and two tanks are provided for this purpose—a suction tank and a discharge tank. The former is located in the basement and is closed or open. The discharge tank may also be closed, in which case it is half full of water and half full of air and the pump is controlled by a pressure regulator. If it is an open tank located on the roof of the building, the pump is controlled by a float.

The elevators take their water from the discharge tank and discharge it back into the suction tank. The water that the pump handles is, therefore, clean and the plunger and ring type is well adapted for all pressures up to 150 pounds per square inch. For higher pressures packed plungers must be used.

120. Fire Pumps.—These pumps must be designed with special consideration as to reliability and durability. The valve area must be extra large as the demand for water may exceed the rating at any time, and the pump can then be run at a very high speed even if the operation be what one would call unsatisfactory in an ordinary pump.

This can be accomplished by selecting the next larger size pump body and fitting it with a smaller plunger. Thus, to make a 16 × 8½ × 12-inch fire pump, a 10¼ × 12-inch pump body would be selected and fitted with an 8½-inch plunger. A point that refers to all other pumps, as well, may be noted here. By reducing the size of the plunger, we save only the difference between the larger steam end and the smaller one. With an outlay of a little more money, it would be possible to buy an 18 × 10¼ × 12-inch pump with exactly the same capacity, but at a moderate speed, as the above 16 × 8½ × 12-inch pump could have at its highest speed. In fact, it is likely that the larger pump would give more water; but the valve area, expressed in percentage, would be too small for a fire pump. This shows how easy it is to arrive at wrong conclusions in calculating pumps where the valve area is considered only, instead of considering it in relation to the piston speed.

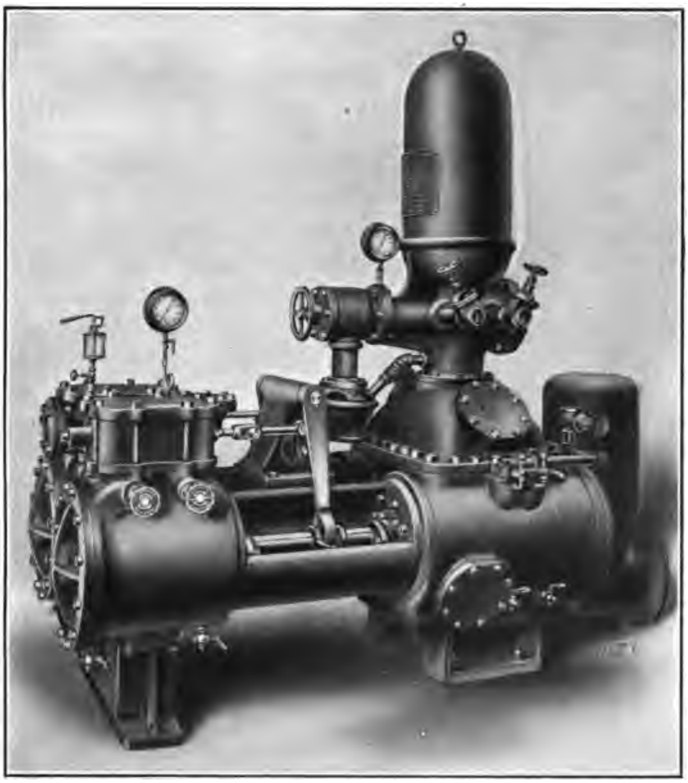

Fig. 188.—Underwriter fire pump.

About 20 years ago, John R. Freeman took hold of the problem of creating a really reliable fire pump. He analyzed the situation carefully and decided on a duplex pump with simple steam cylinders. He selected several sizes and prescribed the necessary fittings so as to make it a rugged, rust-proof and reliable pump, one that any inexperienced man, who may be excited and in a hurry, could start up instantly without doing any damage. Full specifications for this pump have been published by the National Board of Fire Underwriters. It is called "The Underwriter Fire Pump" or "The National Standard Fire Pump," and has many excellent features. The perusal of these specifications can be recommended to anyone desiring to know how a duplex pump should be made. Fig. 188 illustrates a fire pump built and equipped according to these specifications.

121. Waterworks Pumps.—There are three distinct methods of supplying a town with water by means of pumps. The most satisfactory is naturally the use of a large elevated reservoir into which the water is pumped from the water supply. This is located on a hill near the town and furnishes a constant pressure in the main pipe. The pumps then work against a constant head.

In towns that are not so favorably situated, the reservoir is replaced by a large steel standpipe holding a few hours' supply. As the water consumption changes, the level in the standpipe and the pressure in the main also change. The pumps need constant attention so as to adjust them for the varying water supply.

For fire service, if any is desired, the reservoir or standpipe is shut off from the main pipe and the pump delivers directly into it.

The third method is the direct system where the pump delivers directly into the main for domestic as well as fire service. It is then controlled by a pressure regulator which keeps the water pressure constant within narrow limits.

Any type of pump may be used for this purpose.

122. Mine Pumps.—Pumps used for draining mine shafts are usually built along the same lines as the pressure pumps, except that they are larger and are constructed for moderately high pressures. The lifts for which such pumps are built vary from 500 to 1000 feet. Deeper shafts are divided into relays and a "station pump" is installed at approximately every 500 feet measured from the surface.

In sinking a mine, the first pump used is a vertical one that can be lowered down following the receding water level. A single outside, center-packed pump used for this service is illustrated in Figs. 189 and 190. It is fastened to two flat

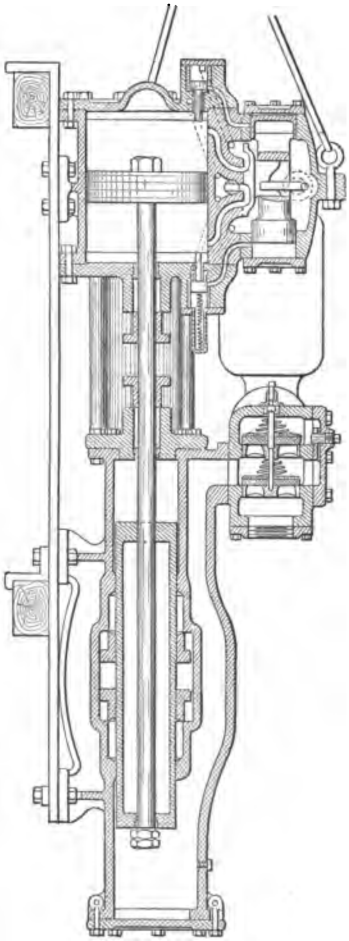

FIG. 189.—Cross-section of single sinking pump.

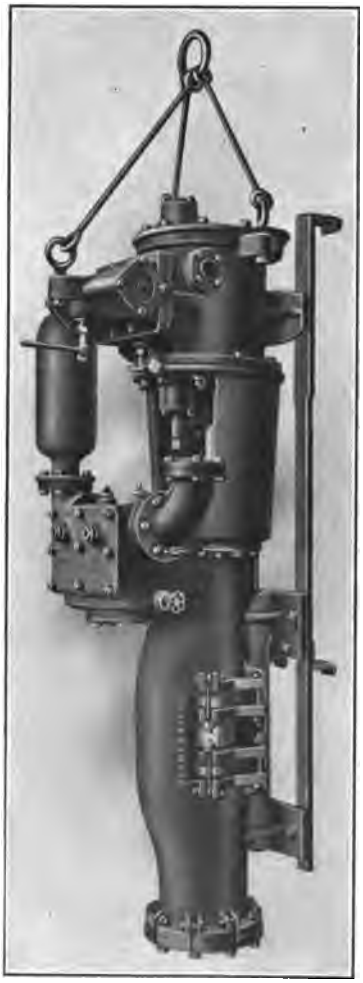

FIG. 190.—Single sinking pump.

strips of steel, the "sinking irons," which are provided with hooks intended to be hooked over a beam. Suitable eye-bolts are provided, by means of which the pump can be conveniently handled. The suction pipe, which is attached to the lower end of the pump body, is made of a piece of rubber hose. Moreover, bolts that require frequent handling are hinged, so as to prevent them from being dropped and lost.

The Cameron type is admirably adapted for this service because it has no external valve motion that may be damaged by flying stones. Single pumps are preferred for this purpose on account of the small space they occupy in the shaft. A duplex pump for this service is shown in Fig. 191.

After a certain depth has been reached, say at 500 feet from the surface, a chamber is cut out and a large stationary pump, called the "station pump," is installed therein. A reservoir, called the "sump," is then provided, into which the sinking pump delivers its water, and from which the station pump lifts it to the surface. The boilers are at the surface, and well-covered pipes convey the steam down the shaft to the pump. The exhaust steam is condensed by a jet condenser and discharged into the sump.

At the 1000-foot level, a second station pump is installed which delivers its water to the "sump" of the first, and so on.

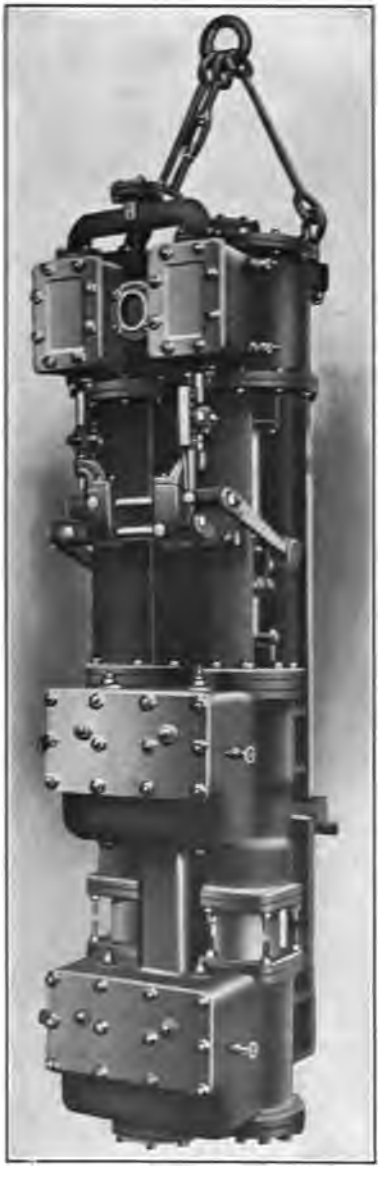

FIG. 191.—Duplex sinking pump.

The principal requirements of a mine pump are reliability and convenience of repair. For this reason, the pump should be sectionalized as far as is practicable, which feature allows it to be easily handled, with the additional advantage that, in case a part is broken or worn out, it can be readily and cheaply renewed. Furthermore, by limiting the size of the various parts, simple castings are obtained which are more reliable than large ones of complicated design and with more or less unequal distribution of metal.

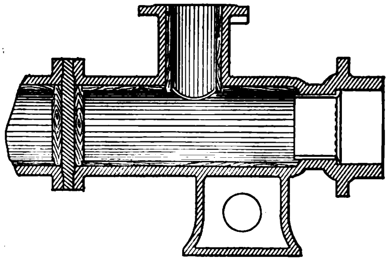

Fig. 192.—Wood-lined pump body.

The importance of this branch of the pump business may be realized if the enormous quantities of water that have to be handled are considered. Thus, in coal mining approximately 15 tons of water are pumped to the surface for every ton of coal taken out.

The water to be pumped out of a coal mine is often highly acidulous, containing in some mines as high as 160 grains of free sulphuric acid per gallon. In such cases, care must be taken to have no iron come in contact with the water. Either the whole pump is made of a copper and tin composition, with a little lead added to make it flow more readily, no zinc being allowed to be present, or it is made of iron and the entire interior lined with lead or wood. Fig. 192 illustrates how a wood lining is applied to the working barrel of a pump. Staves of pine wood are placed

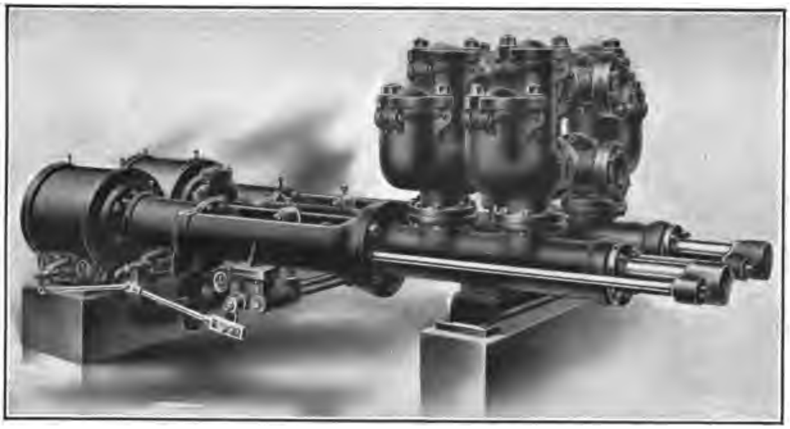

Fig. 193.—Mine pump.

all around the inside of the pump barrel, the last two staves being tapered so as to form wedges. The last stave is then driven home, thus fixing all the others in place. The projecting ends are sawed off flush with the face of the flange and then the holes in the nozzles are sawed out. Parts that cannot well be wood-lined, such as valve chambers, are covered with antimony lead. A wood-lined pump, after having once been in service, should not be allowed to dry out as this would ruin the lining.

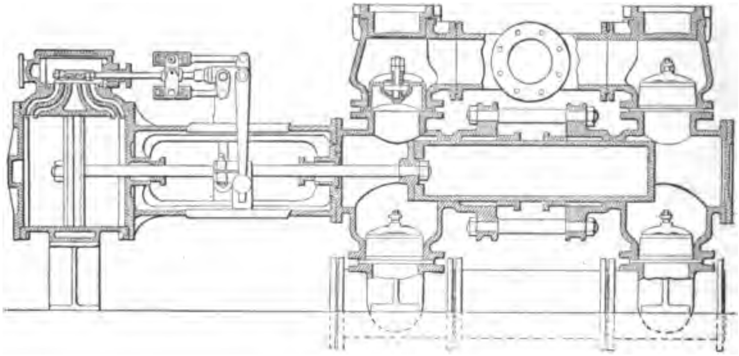

FIG. 194.—Mine pump with annular valves.

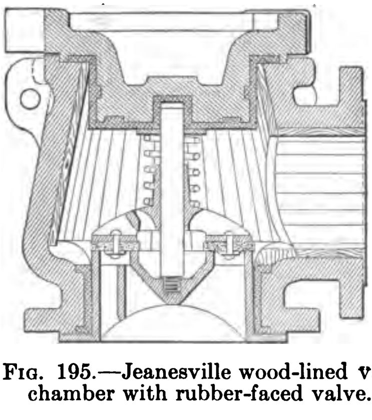

FIG. 195.—Jeanesville wood-lined valve chamber with rubber-faced valve.

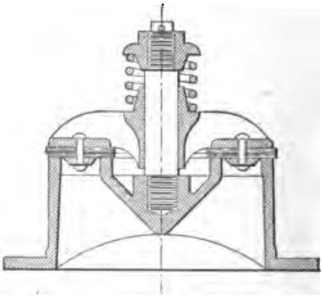

FIG. 196.—Leather-faced annular valve.

For ordinary mine service, the outside center-packed plunger pump with sectionalized pump end is very serviceable.

The pump illustrated in Fig. 193 is especially designed for extra deep mine work, and is built after the pressure-pump type. There is only one pot in each corner and each contains a number of small valves. The covers are held on by hinged bolts which cannot be lost, and thus easy and convenient access can be had to the valves.

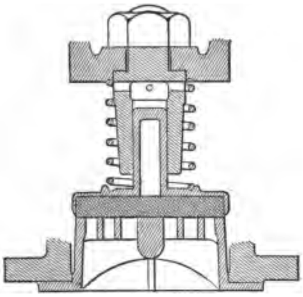

FIG. 197.—Solid rubber valve.

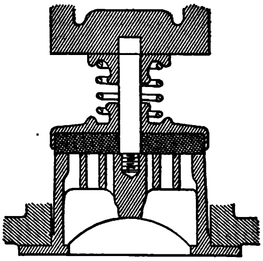

FIG. 198.—Perforated rubber valve.

Fig. 194 shows a mine pump that was developed for service in the anthracite coal field. Here large annular valves are used, one in each corner. They may be either rubber or leather faced, as shown in Figs. 195 and 196 respectively.

In another type used extensively in the anthracite coal fields, the valve is a thick rubber disk seating on a strongly ribbed valve seat. Two forms of valves are used, as may be preferred by the customer. One is a solid rubber disk backed by a bronze casing guided in the cover, while the other is a rubber disk with a hole in the center and guided on a central stem fastened to the valve seat. These two types are illustrated in Figs. 197 and 198.

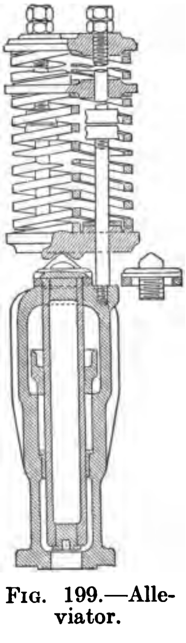

FIG. 199.—Alleviator.

123. Alleviators.—On pumps where the water pressure exceeds 300 pounds air chambers cannot be used, because the water rapidly absorbs any air with which the air chamber may be supplied. If an elastic element is desired, an "alleviator" may be used. Such an apparatus is illustrated in Fig. 199, and is seen to consist of a spring-loaded ram which yields to any excess of pressure and thus relieves the pump end of any undue stresses.

124. Accumulator Pumps.—Accumulators are used chiefly with hydraulic presses and their object, as generally stated, is to supply momentary demands for water, which the pump can then replenish during the period of idleness of the press. Therefore, only a small pump is required, the size of which is calculated for the average capacity instead of the maximum capacity. This, however, in most cases, is not the principal reason, because the cost of the accumulator is much higher than the additional price of a larger pump. The purpose is rather to have an elastic element in the pipe which will take up all irregularities in the demand for water, act as a cushion against shocks and give the pump time to start slowly and quietly.

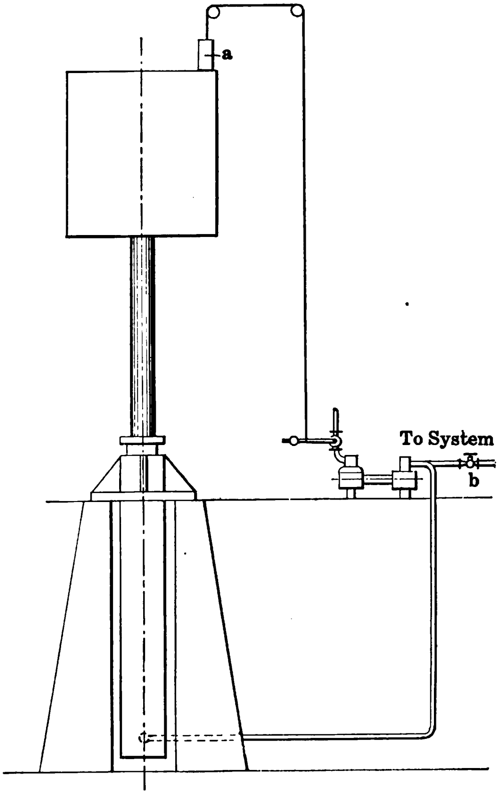

FIG. 200.—Weighted accumulator and pump.

In connection with direct-acting pumps two types are used—the weighted accumulator and the steam accumulator. The former consists of a hydraulic cylinder, the upper end provided with a stuffing-box through which a ram works. This ram carries a platen loaded with a heavy weight, which is equivalent to the area of the ram multiplied by the water pressure. In operation the pump will always tend to keep the ram in its highest position and must stop when this position is reached. For this purpose the small weight *a*, Fig. 200, is suspended from a rope attached to the lever of the chronometer valve controlling the steam supply. This lever is itself weighted and descends when the ram lifts the weight *a*, thus shutting off the steam supply and stopping the pump. When the valve *b*, in the pipe system, is opened for the purpose of starting one of the hydraulic presses, the ram will instantly drop and supply this sudden demand; the weight *a* will then descend and start the pump. If it were not for the accumulator the pressure would drop considerably during the time it takes to start the pump, which may require from 10 to 20 seconds. The displacement of the

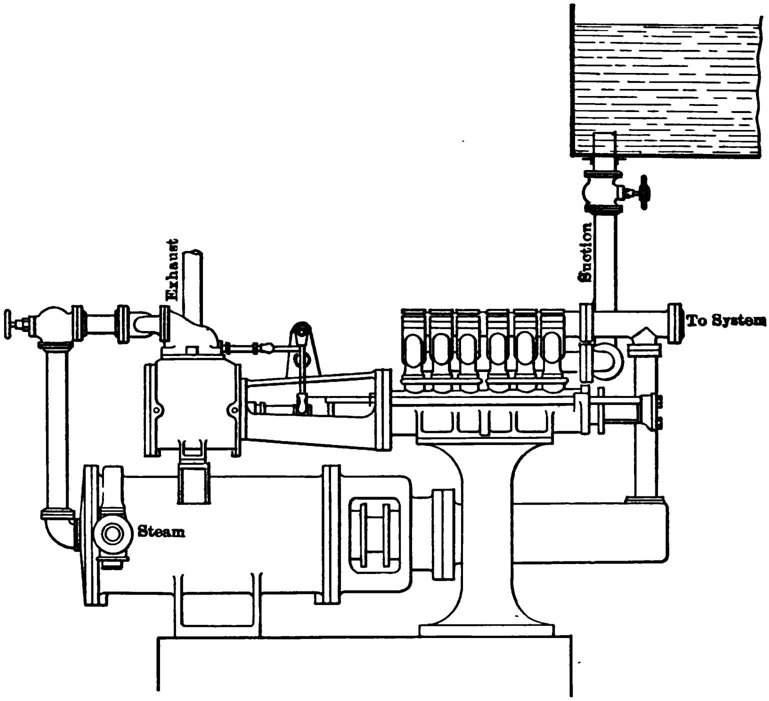

Fig. 201.—Horizontal steam accumulator and pump.

ram is, therefore, generally made equal to the maximum capacity of the plant in 20 to 30 seconds, and the pump will have that much time to get under way.

Steam-driven accumulators are the invention of Mr. C. C. Worthington. Here a steam-loaded piston is used instead of the dead weight, and provides a much more elastic and safer element. Shocks are entirely eliminated, since the weight of the ram and piston is small and no great inertia has to be overcome in reversing its motion, as is the case in the weighted type.

Fig. 201 shows a steam accumulator with pump and connections, and at the same time brings out the important advantage that it may be placed in a horizontal position. Fig. 202 shows the steam accumulator in section. The steam piston *a* and the ram *b* are rigidly connected and each is fitted to its cylinder. The steam supply to the pump is regulated to meet the variations of the water consumption in the following manner:

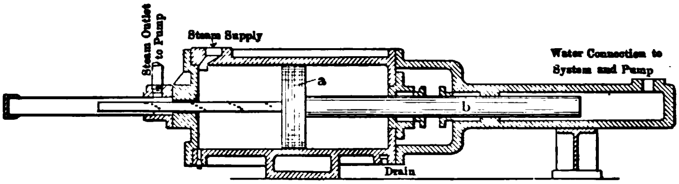

FIG. 202.—Section through steam accumulator.

The steam pressure in the steam cylinder is maintained constant by a reducing valve located in the steam pipe just ahead of the cylinder. This valve is set for a drop of about 5 per cent. and its chief object is to provide some means of regulating the water pressure. The steam supply to the pump is taken from the interior of the steam cylinder through a perforated pipe which is either attached to the steam piston and works through a sleeve in the head, or, if the ram is large enough, may be attached to the head and work through a sleeve in the steam piston, extending into the ram. In the latter case no pipe extends from the steam cylinder head and the whole apparatus becomes much shorter. The normal position of the piston is close to the head when all the holes in the pipe are closed and the pump is at rest. A sudden demand for water will cause the piston to descend or move away from the head, and in doing so it will uncover some of the holes and steam will begin to flow through them to the steam chest of the pump. The pump will then start and in a fraction of a minute will have attained full speed. It will then supply the demand for water and in addition will replenish that taken from the hydraulic cylinder and bring the ram back to its normal position.

The system being closed in itself requires that the sizes of the pump and the accumulator be calculated with great care. The principal point to be considered is that the steam cylinder of the pump be made large enough. The following rules should be observed:

Displacement of ram = Capacity of plant in 30 seconds.

Capacity of pump = $1\frac{1}{2} \times$ capacity of plant.

Steam pressure in accumulator = Steam pressure in pipe $\times$ 0.95.

Steam pressure in pump = Steam pressure in accumulator $\times$ 0.95.

The following example will explain this:

Capacity of plant.................... = 40 gal. per min.
Water pressure....................... = 2000 lb. per sq. in
Steam pressure....................... = 120 lb. per sq. in.

Then according to the foregoing rules:

Displacement of ram.................. = 20 gal.
Capacity of pump..................... = 60 gal. per min.

Assume a duplex pump of 12-inch stroke, a piston speed of 72 feet per minute and 65 per cent. mechanical efficiency.

The diameter of the plunger is then 2¾ inches, and the accumulator water end will be 9 inches diameter by 72 inches stroke, which gives a displacement of nearly 20 gallons.

The steam pressure in the steam cylinder of the accumulator will be $120 \times 0.95 = 114$ lb., and the ram load is $63.62 \times 2000 = 127{,}240$ lb. The area of the steam piston will then be

$$\frac{127240}{114} = 1120 \text{ sq. in.}$$

which corresponds to a 37.8-inch diameter. One would, therefore, select a 38-inch piston and correct the steam pressure to 112 pounds.

This pressure is reduced 5 per cent. in its travel to the pump, giving a steam pressure in the steam chest of 106 pounds. The mean effective pressure is then[1] $106 + 10 - 16 = 100$ lb. per sq. in. The plunger load equals the area of the plunger times the water pressure, or

$$5.94 \times 2000 = 11{,}880 \text{ lb.}$$

and the steam force is

$$\frac{11880}{0.65} = 18{,}400 \text{ lb.}$$

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[1] See page 108.

With a mean effective pressure of 100 there would be required 184 square inches in the steam cylinder, corresponding to a diameter of 15.3 inches. Therefore, a 16-inch cylinder would be selected. Fig. 203 shows the calculated results diagrammatically.

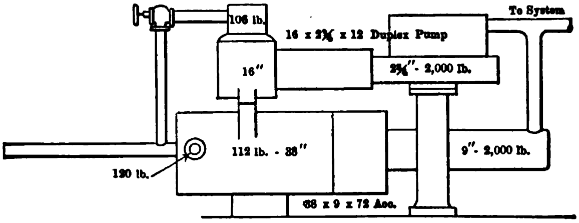


FIG. 203.—Calculation of steam accumulator.

125. Oil Line Pumps.—When oil was discovered in Pennsylvania in large quantities, the problem of transportation was solved by laying pipe lines along the ground and pumping the oil from one station to another. For this service direct-acting pumps were well adapted and exclusively used. The Pennsylvania oil, being light and of paraffine base, did not present any greater difficulties in pumping than water. The following formulas may be used for this work:

$$p = \frac{9B^2}{d^5} \tag{1}$$

$$B = \sqrt{\frac{d^5p}{9}} \tag{2}$$

where p = Friction in pounds per square inch for 10 English miles;
d = Diameter of pipe in inches;
B = Barrels per hour at 38 deg. Baumé.

In formula (1), for every 3 deg. above 38 deg. Baumé subtract 2 per cent. from p and for every 3 deg. below 38 deg. Baumé add 2 per cent. to p.

In formula (2), for every 3 deg. above 38 deg. Baumé add 1 per cent. to B and for every 3 deg. below 38 deg. Baumé subtract 1 per cent. from B.

Greater difficulties were experienced with the Texas and California oils which are of asphaltum base and very viscous. These oils offer an enormous resistance to being pumped through a pipe and will not move at all at low temperatures. It therefore becomes necessary in such cases to heat the oil with the exhaust steam from the pump.

In 1902 A. F. L. Bell made some tests in the Kern oil fields with heavy California oil. The pipe diameter was 8 inches and the following formula agrees well with the data obtained:

$$p = \frac{32768}{d^5} \times \frac{9}{t - 40} \sqrt[3]{B^4}$$

where d = Diameter of pipe in inches, for 1 mile;
p = Friction in pounds per square inch;
t = Temperature of oil in deg. F.;
B = Barrels per hour.

The gravity was 14½ Baumé at 60 deg. F.

It is assumed that the friction varies inversely as the 5th power of d, no data being available for pipes of other diameters. The second term of the expression was developed on the assumption that at 40 deg. F. the oil ceases to flow. The capacity then becomes zero.

Tables 25 to 29, given in the appendix, were calculated from the above formulas. It will be noted that while the table for Pennsylvania oil is calculated on a basis of 10 English miles and can be used for any length by simply multiplying the values given by one-tenth of the length, the table for California oil is given for 1 mile only. The reason for this is that the oil cannot retain its temperature for any length of time but has a different temperature at every point. This change of temperature is very difficult to predict and impossible to generalize, because it depends upon the manner in which the pipe is laid and the temperature of the outside air.

The two sets of tables may be taken as the limits, the former being correct for the very lightest oil and the latter for the very heaviest California oil. They should not be used, except as a guide, for oils of intermediate viscosities.

126. Vertical Pumps.—The simplest design of a vertical pump is that of a bucket pump. A "bucket" is a piston provided with pump valves which open on the down stroke and allow the water and air to pass through to the upper end of the cylinder. On the down stroke the piston is therefore balanced and is merely descending through the water without doing any useful work. On the up stroke the valves close and the air and water are discharged through the discharge valves, located in a valve deck in the upper end of the cylinder. Access to the bucket valves is gained through a handhole in the cylinder wall; see Fig. 204*a*. Although, theoretically, a third set of valves is not needed for the operation of the pump, it has been found advantageous to place another deck in the lower end of the cylinder acting as a foot valve and assisting greatly in maintaining the high vacuum under which these pumps are intended to work in their capacity as wet vacuum pumps.

As the pumps are single-acting, two are generally connected together by means of a beam, as shown in Figs. 204*b* and 205, making the whole outfit equivalent to a double-acting pump.

The periphery of the bucket is provided with a space for fibrous packing, held in place in the usual manner by a gland.

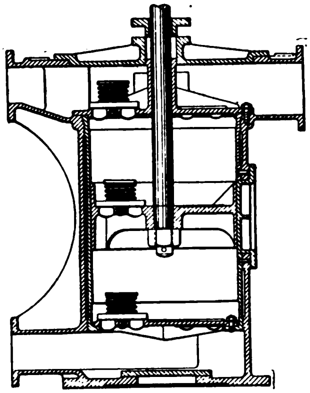


FIG. 204*a*.—Section through vertical bucket pump body.

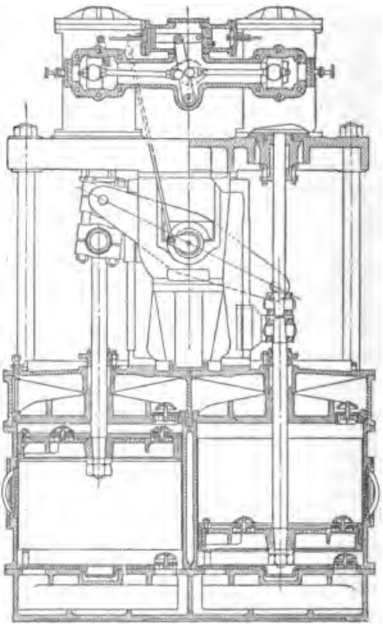


FIG. 204*b*.—Section through vertical wet-vacuum pump driven by duplex steam end.

The design next to this in simplicity is the outside-packed, single-acting plunger pump, such as is used in connection with triple-expansion, crank-and-flywheel pumps and triplex power pumps. It is hardly ever used with a direct-acting steam end and, therefore, need not be considered here, except as an intermediate step to the more complicated designs.

There are two possible ways of designing this type of pump: The discharge valves may be arranged around the plunger and the suction valves in a deck below the plunger, as in Fig. 206; or the valves may be placed in a valve chamber that is separate from the pump body and connected to it by pipes. The differential plunger pump, illustrated in Fig. 207, is similar to this and is well adapted to the direct-acting principle. Here the valves are arranged in a similar manner, but the discharge chamber *a* is extended upward so as to embrace the plunger *b*, the piston rod being enlarged to a diameter having exactly one-half the area of the plunger *b*. It thus forms the small plunger *c*, extending through a stuffing-box in the upper end of the cylinder. It is readily seen that the suction is single-acting, while the discharge is double-acting. The single-acting suction causes a slight difference in the plunger loads between the up and down strokes, which can easily be taken care of, together with the

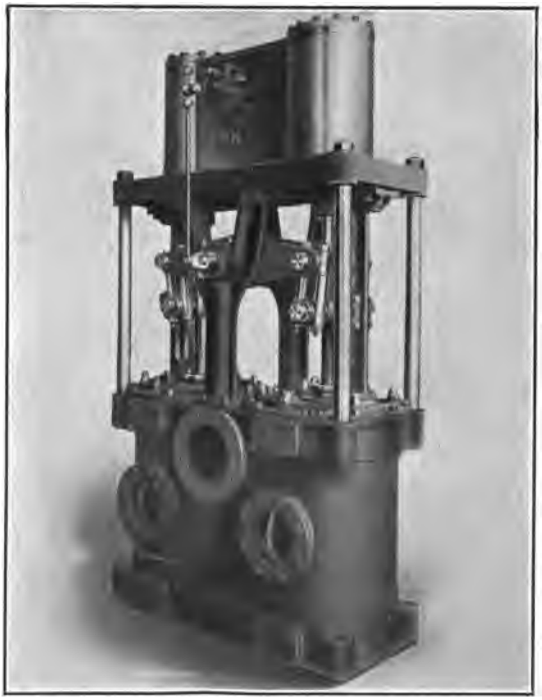


Fig. 205.—Vertical wet-vacuum pump, twin pattern.

weight of the reciprocating parts, by a balancing device which will be explained later.

Figs. 209 and 210 illustrate two designs of vertical double-acting piston pumps. The former is a low-service type and is used as a ballast pump or for tank service. There are two valve decks each partitioned off so as to form four valve chambers. That in Fig. 210 is of the heavier pattern, which is indicated by the absence of flat walls. The valves are of the flat-seated wing type and are accessible through individual covers.

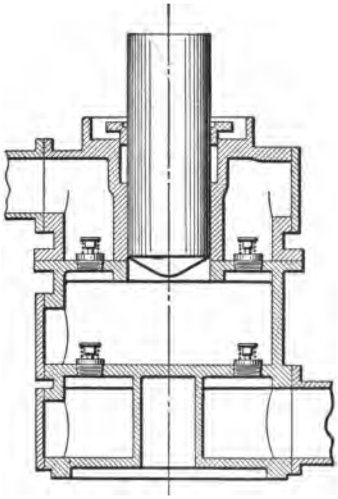


Fig. 206.—Vertical single-acting pump.

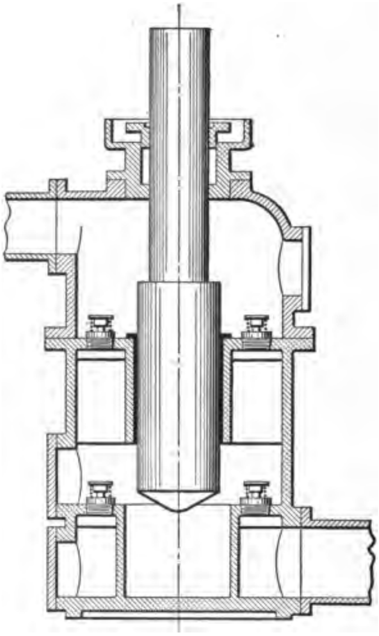


Fig. 207.—Vertical differential plunger pump.

Larger vertical, double-acting pumps are more difficult to design since there are four valve decks, arranged above each other with two suction chambers and two discharge chambers in between, all connected together by communicating pipes. In addition, space must be provided for numerous handholes for access to the valves; see Fig. 208. For high pressures and very large plungers the pump body may be separated from the valve chamber, as in Figs. 212 and 213.

127. Balancing Devices.—A direct-acting pump cannot work properly unless the forces are accurately balanced, as there is no heavy flywheel to take up any difference between the force and the load. In a vertical pump, the weight of the reciprocating parts causes a difference in the acting forces which tends to accelerate the speed on the down stroke and retard the speed on the up stroke. For the sake of uniform delivery it is essen-

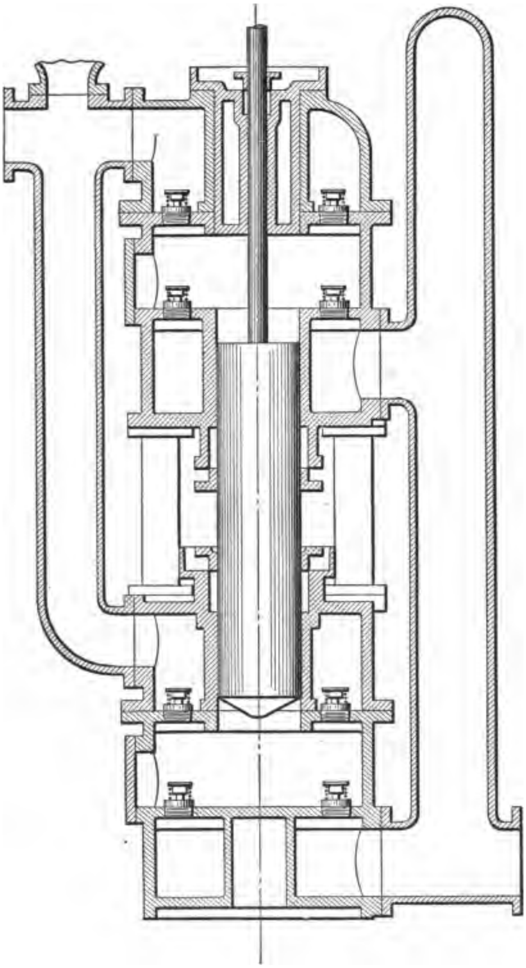


Fig. 208.—Vertical outside center-packed plunger pump.

tial that the speed be uniform and several devices are in use for accomplishing this purpose.

In small pumps, having a stroke of from 12 to 18 inches, a choke valve placed in the lower exhaust port will answer the purpose. This choke valve is closed until the back pressure created thereby under the cylinder produces the required retardation in the down stroke.

The same device may be used on compound close-connected vertical pumps. Here the choke valve is placed in the upper low-pressure steam port. It then serves the double purpose of creating a back pressure on the under side of the high-pressure piston and reducing the pressure on top of the low-pressure piston. Fig. 211 shows this device. It also shows the way the valve rods are balanced, both being extended through stuffing-boxes at the upper end, of the same size as the lower ones. This is important since the valve rod is driven by a lost-motion link and the steam pressure acting on the unbalanced area of the rod together with the weight of the rod, slide valve

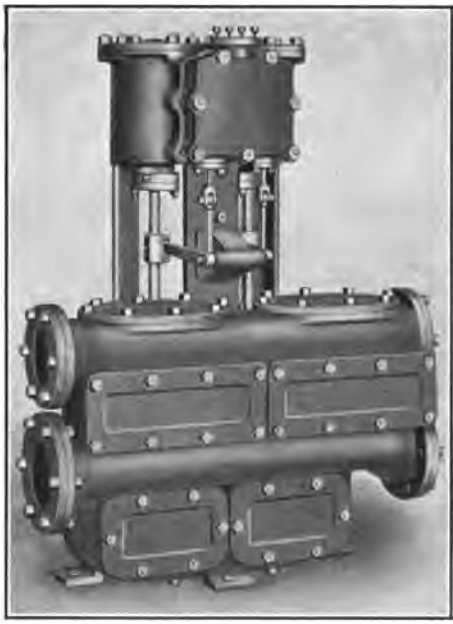


Fig. 209.—Vertical ballast pump.

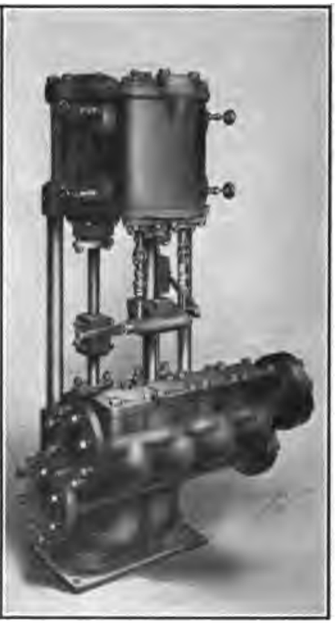


Fig. 210.—Vertical boiler-feed pump.

and link might be sufficient to overcome the friction in the stuffing-boxes and cause the rod to drop.

If the compound pump is not close connected, that is, if it is located in a deep pit, with extended rods between the steam and the water ends, a steam balancing device is used as well as on all triple-expansion pumps of 18-inch stroke. This consists of an auxiliary steam cylinder mounted on top of the high-pressure cylinder which has its lower end connected to the steam supply pipe; see Fig. 212. The two balancing cylinders are connected together by communicating pipes, which gives a fairly constant pressure under the balancing pistons. This is used on all compound pumps of 24-inch stroke.

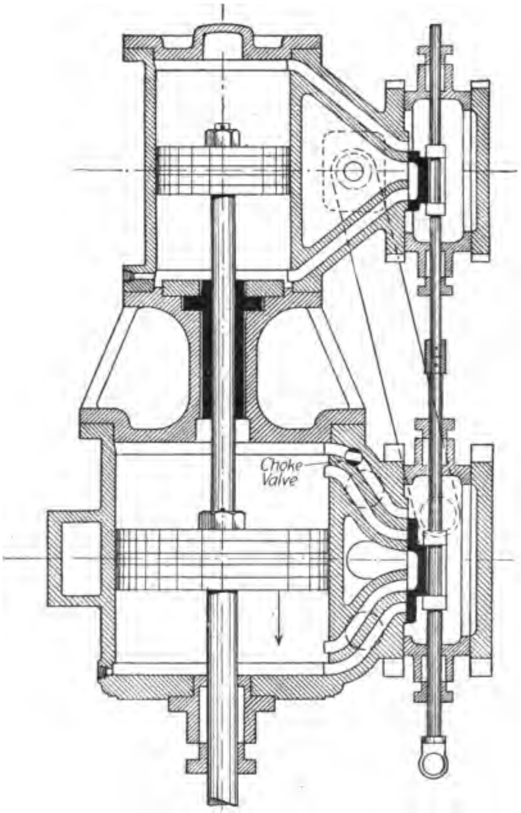


Fig. 211.—Vertical compound steam end balanced by choke valve.

Triple-expansion pumps of 24-inch stroke, 36-inch compound pumps and all larger sizes should be provided with an hydraulic balancing device, such as illustrated in Fig. 213. This consists of a plunger *a* attached to the piston rod at some convenient place and working in an hydraulic cylinder *b*, the lower end of which is connected to a tank *c*, partly filled with water under pressure. This tank should be large enough so that the variation of pressure does not exceed 2½ per cent., which requires an air space of

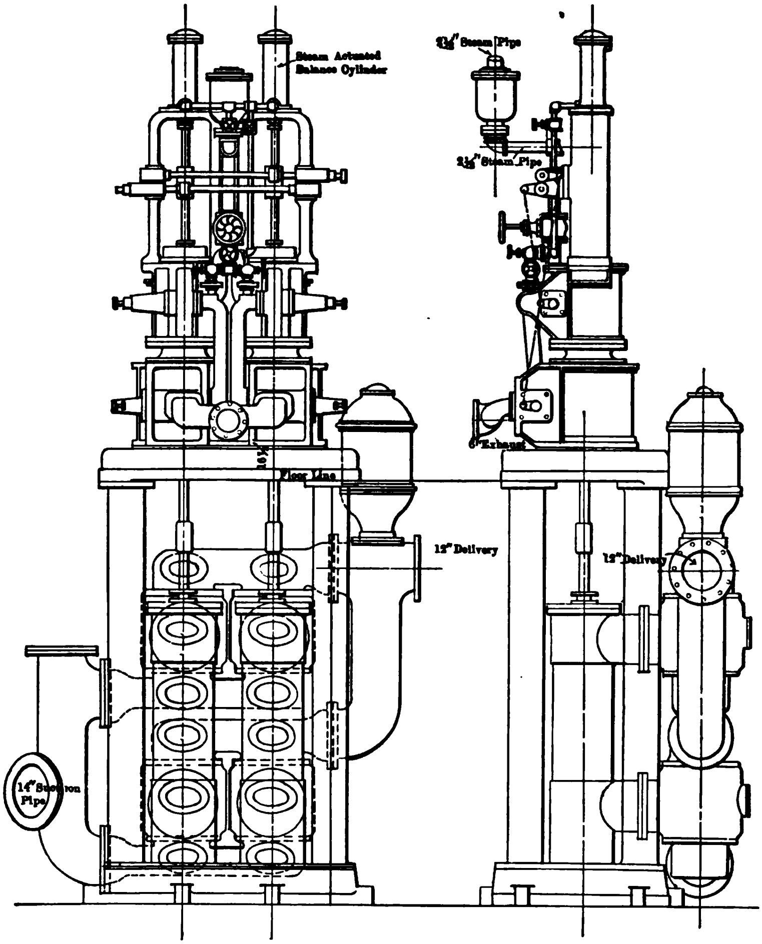


FIG. 212.—Vertical double-acting piston pump with small separate valve chambers and steam balancing device.

forty times the displacement of one balancing plunger. If the size of the plunger is calculated for a water pressure about 10 per cent. less than the discharge pressure, the tank can be filled from the discharge pipe and does not require any extra pump.

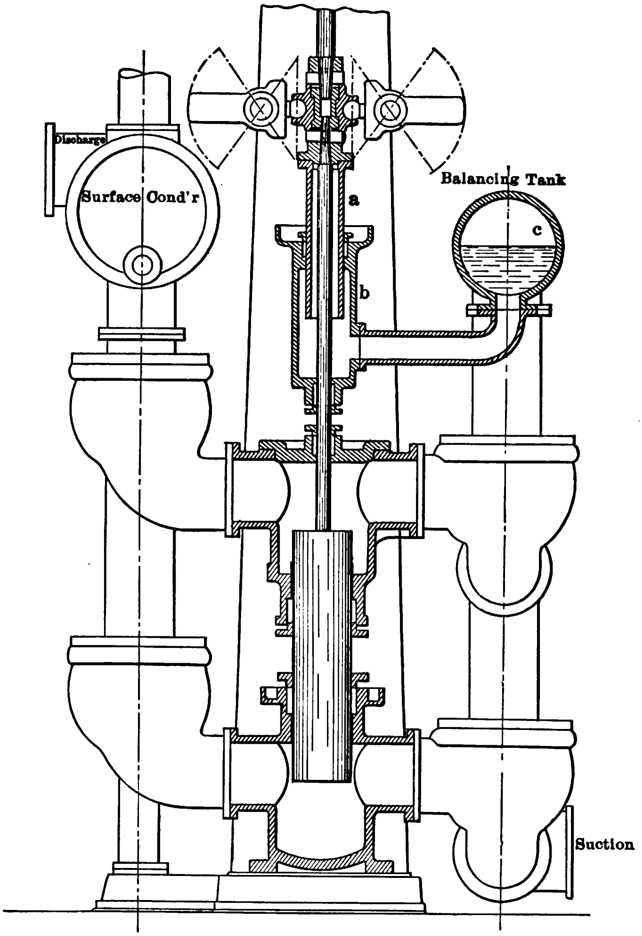


Fig. 213.—Vertical outside center-packed pump with hydraulic balancing device.

128. Deep-well pumps of the bucket type are designed for use on nonflowing wells where the water does not stand within suction distance. Two methods are employed, namely: the *plain-tube* well and the well with a *drop pipe*. In the former, shown in Fig. 214, the pipe with open ends is sunk to the proper depth where a sufficient water-bearing strata is penetrated. Strainer *A* is then lowered to the bottom of the well and the pipe is drawn back far enough to expose the slotted portion. After this the working barrel *B* is lowered into the gum-packer on top of the strainer and is tapped firmly into place. The bucket *C* is driven by means of wooden rods of convenient length (18 feet)

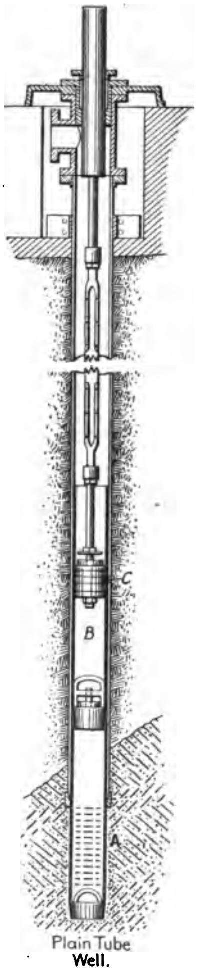


Fig. 214.

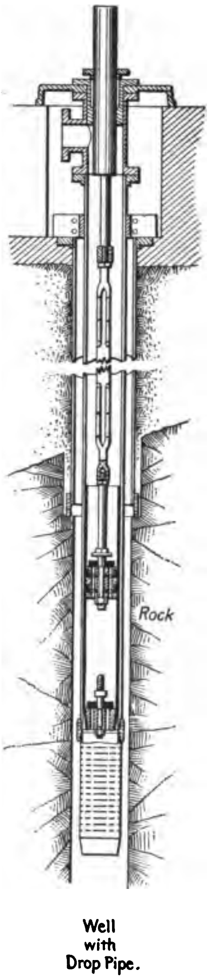


Fig. 215.

Deep-well pumps.

coupled together. The top of the well pipe is provided with a tee for a discharge connection and has a stuffing-box through which a brass plunger works.

If the well discharges at the surface this plunger is made one-half the area of the bucket, resulting in a uniform discharge. If the pump discharges into an elevated tank the top plunger is made larger so as to throw the effect of the additional head upon the plunger and thereby assist in the up stroke.

The well with a drop pipe (Fig. 215) differs from the plain-tube well in that the working barrel is attached to a pipe and is lowered into the well pipe. This is a better arrangement in case of a small supply, as it allows a head of water to collect above the bucket during the down stroke. It is used in wells that are either wholly or partly drilled into rock and in wells where the casing does not extend low enough to reach the water supply; also in old wells with defective or bent casing.

In some cases the drop pipe is extended to the bottom of the well and the bottom length is perforated to act as a strainer. In this event the working barrel is placed just far enough above the bottom to be within easy suction range, but removed from any dirt which may get into the bottom lengths. When this plan is used it is not necessary to place a clamp at the top, as the pipe resting on the bottom of the well supports its own weight.

CHAPTER VII

DUTY

129. Duty.—Steam engines are usually tested for economy by determining the steam consumption per indicated or brake horsepower per hour. In testing a pump, however, its *duty* is generally determined.

Duty is the number of foot-pounds of useful work done by one thousand pounds of dry steam.

Originally, duty meant the amount of work done by 100 pounds of coal, since this was the article the customer was chiefly interested in saving, but as this involved the quality of coal and the efficiency of the boiler, which was rarely furnished by the pump manufacturers, this definition soon fell into disuse.

A high-grade boiler can evaporate 10 pounds of water per pound of coal. Therefore, in order to make a fair comparison, $10 \times 100 = 1000$ pounds of steam was adopted as a basis for rating the economy of pumps. The two methods are closely related and one can easily be converted into the other.

Let F = Pounds of steam (or feed water) used per horsepower per hour;

and

D = Duty in foot-pounds per 1000 pounds of steam.

Then,

Steam consumption = F lb. per hr. for 33,000 ft.-lb. per min.

Then,

$$\frac{60 \times 33000}{F} \text{ ft.-lb. per min. require 1 lb. of steam per min.}$$

and by multiplying by 1000, and eliminating the time,

$$\frac{60 \times 33000 \times 1000}{F} \text{ ft.-lb. require 1000 lb. of steam.}$$

This is, therefore, the duty (D), and

$$D \times F = 1{,}980{,}000{,}000 \text{ ft.-lb.}$$

This formula enables one to readily calculate the rating in feed water or steam consumption if the duty is given and *vice versa.* Thus, a duty of 100 million foot-pounds per 1000 pounds of steam is equivalent to 19.8 pounds of steam per horsepower-hour.

A committee of the American Society of Mechanical Engineers reported in 1891 on a standard method of conducting duty trials.

Instead of the above units of 100 pounds of coal, or 1000 pounds of steam, they recommended a new unit, namely: one million B.t.u. furnished by the boiler. The economy would then be expressed in foot-pounds of work done per one million B.t.u. This unit is the equivalent of 100 pounds of coal when each pound imparts 10,000 B.t.u. to the water in the boiler.

With a boiler pressure of 150 pounds per square inch and a feed water temperature of 150 deg. F., let it be desired to find out how much water can be evaporated with this heat. Consulting the steam tables (Marks and Davis), the total heat in a pound of steam under these conditions is found to be 1193.4 B.t.u. from 32 deg. F. and 1225.4 B.t.u. from 0 deg. F. The heat imparted to 1 pound of water, then, is $1225.4 - 150 = 1075.4$ B.t.u., and one million B.t.u. would evaporate $1{,}000{,}000 \div 1075.4 = 930$ pounds of water, as against the old unit of 1000 pounds. The duty expressed in the new unit would therefore only be 93 per cent. of that expressed in the old unit, and this fact made the pump manufacturers hesitate to adopt the new unit. It is only natural that every manufacturer wishes to make as good a showing and also create as good an impression as possible for his product, and the one that uses the new unit finds himself at a disadvantage as compared to the one who adheres to the old unit, and has to recalculate his duty and offer explanations.

In Kent's Pocket Book may be found a reprint of the report of this committee.

130. Measuring Duty.—The two principal data required for determining the duty are: the work done (W) and the quantity of steam consumed (S). The first item may be tested in various ways, one of which is to ascertain the capacity and the head.

(a) CAPACITY.—The delivery or the water discharged by the pump can be measured by means of a pitot tube, a venturi meter, or any other meter that is adapted to large flows; or it can be measured by discharging over a weir. However, to quote C. A. Hague, "It is generally conceded by disinterested people that in a properly designed and constructed displacement pump, the plunger displacement will give closer results, in a matter where perfection is impossible, than any other method."

Aside from the slight leakage and loss inherent in the best of pumps, the pump itself is a perfect meter. It is not difficult to determine the amount of loss or leakage, if in the interest of the buyer this is desired; but as such losses, in a well-made pump that is in good order, are very small, there is little occasion to spend money for such information. The leakage in a new pump is seldom more than 3 per cent., so the most practical way to meet this question is to calculate the theoretical plunger displacement, deduct from this the displacement of the plunger rod, and then deduct 3 per cent. for slip.

The diameter of the plunger should be checked by actual measurement. The stroke in a direct-acting pump is variable and must be measured during the test, and the average of the readings must be used in calculating the piston speed.

The rate of pumping thus obtained is then called the *capacity* of the pump, while the amount of water flowing through some metering device is called the *delivery*.

(*b*) HEAD.—The head pumped against is measured by a tested gage on the discharge pipe and to its reading must be added the distance from the center of the gage to the level of the water in the suction well; see Fig. 12. In case the suction pipe is very long, the suction lift is measured by a vacuum gage on the suction pipe close to the pump. The readings of the two gages are added, as is also the distance h_1 between the center of the discharge gage and the point where the suction gage is attached.

The velocity head may, in the case of direct-acting pumps, be neglected since the velocities are low. However, if the conditions are such that the velocity head constitutes a large part of the total head, it must be taken into account. A case may be imagined where it is the sole duty of a pump to move the water in a channel against no head (to induce circulation), or a pump may discharge over a dike and the whole system may form a siphon. Such cases, of course, require special treatment and the manner of testing should be agreed upon beforehand.

The following formula is found in the Report of the American Society of Mechanical Engineers:

$$h = h_d - h_s + h_1 + \frac{V_d^2 - V_s^2}{2g}$$

where h = Total head;
h_d = Discharge head;
h_s = Suction head (which is negative for suction lift);
h_1 = Distance between gages;
V_d = Velocity in discharge pipe;
V_s = Velocity in suction pipe.

The last term is based on the idea that the suction velocity must necessarily be produced by the pump and, therefore, forms part of the work to be charged against the pump. If now the discharge pipe, as is generally the case, is smaller than the suction pipe, it is claimed that this increase in velocity represents energy that the pump has imparted to the water and that it should receive credit for same. In reciprocating pumps, which type produces a high static head and a low velocity, this term is generally of small importance and hardly ever considered. It may assume an important value in centrifugal pumps, which type produces a low static head and a high velocity. Here part of the velocity is later converted into useful head and should, therefore, not be neglected.

If the suction pipe diameter equals the discharge pipe diameter this item cancels out.

The gage reading should be taken in pounds per square inch, and then be converted into feet of water column, for which purpose the temperature in the suction well is noted. This temperature, however, has no influence on the final result since the amount of water pumped, expressed in pounds, is affected in the same ratio. The quantity of water pumped, in pounds, multiplied by the height, in feet, then furnishes the total work done in foot-pounds, or W.

As previously stated, the second item in determining the duty is the quantity of steam used. There are two ways of finding this quantity, namely: by measuring the amount of feed water pumped into the boiler, or by measuring the condensate discharged by the air pump. The former must be used when the steam is condensed in a jet condenser, where it mingles with the injection water before it can be measured; while the latter is employed where a surface condenser is used, and where the engine is supplied with steam from a battery of boilers that supply other machinery as well.

The water pumped into a boiler may be measured by a meter and the water coming from a condenser may be measured on a pair of scales.

Excellent books have been written on this subject of testing and the student is referred to them and also to the codes for testing pumping engines (1890) and for testing steam engines (1903) of the American Society of Mechanical Engineers.

Having determined these two items, it is then easy to ascertain the duty according to the formula:

$$\text{Duty} = 1000\frac{W}{S}$$

131. Estimated Duty.—All this is not difficult when one is dealing with facts obtained by test, but the problem assumes a more difficult aspect when the duty of an engine is to be estimated beforehand and often before it is designed. While, in such cases, one may be guided by the performance of similar engines, he should, nevertheless, be able to approximate the anticipated duty. This can be accomplished by an inspection of the indicator diagram, showing the mean effective pressure, which is a measure of the work done, and also the terminal pressure of the steam, the density of which is a measure of the steam consumed.

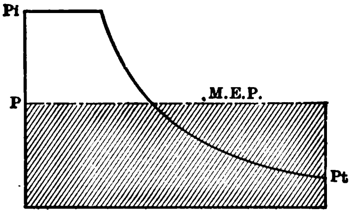


Fig. 216.—Indicator diagram of cut-off engine.

The character of the diagram is immaterial; hence a simple cut-off diagram, as in Fig. 216, will be selected.

Let p = Mean effective pressure in pounds per square inch;
V = Specific volume at terminal pressure, p_t, in cubic feet per pound;
D = Density $\left(\frac{1}{V}\right)$ in pounds per cubic foot;
a = Area of piston in square inches.
L = Length of stroke in feet.

Then the steam force, $F = ap$ (pounds), and the work done per stroke is represented by the area of the shaded part of the diagram, if F is substituted for p.

$$W = FL = apL \text{ (foot-pounds)}$$

The amount of steam accounted for by the indicator at the terminal pressure is:

$$S = \frac{aL}{144} \text{ (cu. ft.)}$$

and, since 1 cubic foot weighs D pounds,

$$S = \frac{aLD}{144} \text{ (lb.)}$$

or

$$S = \frac{aL}{144V} \text{ (lb.)}$$

Hence, the amount of work done per pound of steam becomes

$$\frac{W}{S} = apL\frac{144V}{aL} = 144pV$$

But the theoretical duty is $D = 1000\frac{W}{S}$; hence the card duty = $144{,}000pV$ (foot-pounds per 1000 pounds of steam).

This is called the *card duty*, because it is based on the data furnished by the indicator card.

For the sake of illustration, assume the diagram of a triple-expansion low-duty engine with cross exhausts open, which, as previously shown, consists of three plain rectangles. Let the size of the steam end be 11 & 18 & 29 × 18 inches, which, with an allowance for piston rods, gives cylinder ratios of approximately 1 : 2.6 : 7. Let the boiler pressure be 120 pounds gage which is equivalent to 130 pounds absolute initial pressure at the engine.

According to the formula developed under "Triple-expansion Pumping Engines" (section 66),

$$\text{Mean Effective Pressure} = 3p - \frac{2p}{\sqrt{R_1}} - R_1b$$

where p = Absolute initial pressure;
R_1 = Cylinder ratio of low-pressure cylinder;
b = Absolute back pressure.

Applied to the present example,

$$\text{m.e.p.} = 390 - \frac{260}{2.65} - (7 \times 5)$$

$$= 257 \text{ lb. per sq. in.}$$

If the steam accounted for by the diagram is calculated for various points, a different amount is found for every point and the values will differ considerably. This, of course, would not be the case were the diagram constructed from the steam tables, which would assume that the steam in expanding followed the saturation curve. Nothing, however, would be gained by it, as the steam consumption can be taken from any one point and that which would be given by any other point can be disregarded. The point that has been found the best on which to base this calculation is the terminal pressure of the high-pressure cylinder. To the steam consumption so found, certain losses must be added, and if any other point were selected it would simply mean that different allowances would have to be made in order to arrive at the same result. In the case in question, the terminal pressure of the high-pressure cylinder is 130 pounds absolute. The density at this pressure (as per Marks and Davis Tables) is $D = 0.2897$, and the specific volume is $V = 3.452$, which would make:

$$\begin{aligned} \textit{Card Duty} &= 144{,}000 \times p \times V \\ &= 144{,}000 \times 257 \times 3.452 = 128 \textit{ million foot-pounds.} \\ &\quad \textit{per } 1000 \textit{ lb. of dry steam.} \end{aligned}$$

From this, the unavoidable losses must be deducted, namely: friction loss and steam loss. No rules can be given for these, as they can be found only by experience.

The "Duty Tables" (see pages 232 to 238) give fair values of the steam losses that may be anticipated, and also the mechanical efficiencies. For a triple-expansion condensing pump of 18-inch stroke we find a mechanical efficiency of 82.5 per cent. and a steam loss of 25 per cent., which expressed in the term "steam efficiency" is 75 per cent. Therefore, the net efficiency is:

$$0.825 \times 0.75 = 0.62 \textit{ or } 62 \textit{ per cent.}$$

The actual duty at slow speed and with cross exausts open then would be:

$$128 \times 62 = 79 \textit{ million foot-pounds.}$$

In calculating duty, greater accuracy is required than in calculating sizes. In the latter we are guided chiefly by commercial reasons, and always select the next larger commercial size, which will then operate under a reduced initial pressure; hence, it is important to base the economy on this reduced pressure and not on the actual boiler pressure.

The duty guaranteed by the manufacturer is not the everyday duty, but rather the duty that can be obtained during a test with everything in perfect condition and with the pump operating under contract conditions, that is, at high speed and with the cross-exhaust valves closed.

Steam in expanding from one cylinder into the next larger one follows a sloping line as is shown by Fig. 217. This line begins at point C and ends at D, and in order to find the mean pressure, the absolute zero point O of this expansion curve must be located. This can be done graphically, as shown, and the mean pressure found by means of a planimeter or other method. The mean pressure can also be found mathematically as will be explained in the following:

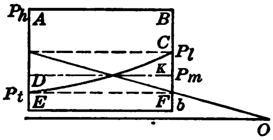


FIG. 217.—Ideal diagram of compound pump with cross exhaust closed.

Let R = Ratio of cylinders;
I = Volume of intermediate space, also expressed in units of volume in terms of the volume of the high-pressure cylinder;
E = Ratio of expansion in this intermediate space (from C to D);
p_h = Absolute initial pressure in high-pressure cylinder at A.
p_l = Absolute initial pressure in low-pressure cylinder at C.
p_t = Absolute terminal pressure at D;
p_m = Absolute mean pressure at K;
b = Absolute back pressure at E and F;
$m.e.p.$ = mean effective pressure.

then

$$p_t = \frac{p_h}{R}$$

and

$$p_l = \frac{p_h\left(1 + \frac{I}{R}\right)}{1 + I} \quad \text{(see section 60)}$$

The ratio of expansion from C to D is

$$E = \frac{p_l}{p_t} = \frac{Rp_l}{p_h} = \frac{R\left(1 + \frac{I}{R}\right)}{1+I}$$

and

$$E = \frac{R + I}{1 + I}$$

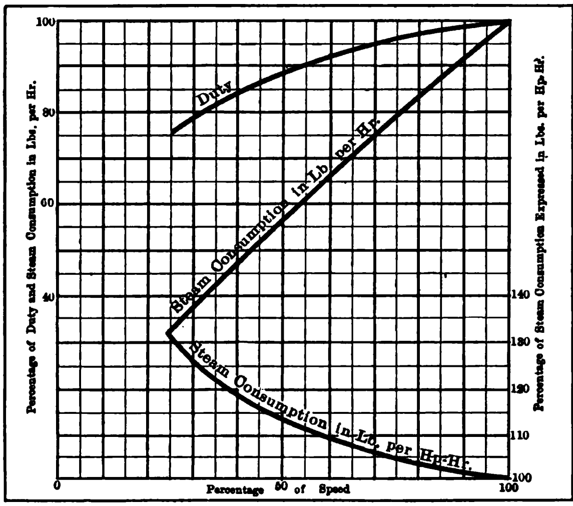
Having determined E we can find the mean pressure by the formula

$$p_m = p_l \frac{\log_e E + 1}{E}$$

and the mean effective pressure in the low-pressure cylinder is $p_m - b$.

The mean back pressure in the high-pressure cylinder will then be $p_m + 2$, allowing a loss of 2 pounds between the cylinders, which makes the mean effective pressure in the high-pressure cylinder $p_h - p_m - 2$. With these two values the steam force of the steam end can be calculated.

The same formulas apply to triple-expansion engines if two cylinders are considered at a time and their mean effective pressures calculated. The mean effective pressures thus found are given in Table 24 marked "Normal Speed, Cross-exhaust Valves Closed" (see page 238), and the duties are calculated from these values.



Load	Duty Per Cent.	Steam Consump. per Hp-Hr. Per Cent.	Steam Consump. per Hour. Per Cent.
Full	100	100	100
¾	96	104.2	78.2
½	88	113.6	56.8
¼	76	131.6	32.9

Full Load is Based on the following Speeds

Stroke In.	R.P.M.	Piston Speed ft. per min.
10	54	90
12	50	100
15	44	110
18	40	120
24	32.5	130
36	25	150
48	20	160
60	17	170

FIG. 218.—Effect of speed reduction on duty.

As is well known, speed has a decided influence on economy. There are losses in the steam end as well as in the pump end that are dependent upon time and if the power developed during this time is reduced, the losses become a greater part of the power and the economy is impaired. Therefore, the duty is always stated for full load (full speed), and for three-fourths or one-half load it is reduced. In Fig. 218 the upper curve gives the duty in per cent. referred to the percentage of speed given on the horizontal reference line. Thus for three-fourths speed the duty is 96 per cent. of that for full speed. Two more curves are added giving the rate of steam consumption in pounds per horsepower-hour and also in pounds per hour.

The above differs from the practice followed in guaranteeing the economy of a steam engine. Here too, guarantees are sometimes made for other than the rated capacity; but there is this difference: The speed of a steam engine is constant and the load may vary, while in a pump the load is usually constant but the water consumption, which governs the speed, may vary within wide limits. The expressions three-fourths load or one-half load in a pump guarantee, therefore, always mean three-fourths or one-half speed.

If the duty is desired for various water pressures, the corresponding steam pressure must be calculated and the duty can then be found by the method explained.

DUTY TABLES

TABLE 18.—SIMPLE-CYLINDER STEAM ENDS

The duties given in these tables are correct only for the mechanical efficiencies stated. For other mechanical efficiencies reduce in proportion

Noncondensing, non-jacketed but lagged. Wire-drawing = 4.7 pounds. Back pressure = 16 pounds

Boiler pressure				50	60	70.0	80	90.0	100	110	120	130.0	140	150.0
Absolute initial pressure				60	70	80.0	90	100.0	110	120	130	140.0	150	160.0
M.e.p.				44	54	64.0	74	84.0	94	104	114	124.0	134	144.0
Card duty, million foot-pounds				45	48	50.5	52	53.5	55	56	57	57.5	58	58.5
Stroke, inches	Mechanical efficiency, per cent.	Steam efficiency, per cent.	Total efficiency, per cent.	Actual duty in million foot-pounds per 1000 pounds of steam										
4	55.0	37.5	21	9.5	10.1	10.6	11.0	11.3	11.6	11.8	12.0	12.1	12.2	12.3
6	65.0	40.0	26	11.7	12.5	13.1	13.5	13.9	14.3	14.6	14.8	15.0	15.1	15.2
8	70.0	42.5	30	13.5	14.4	15.2	15.6	16.1	16.5	16.8	17.1	17.2	17.4	17.6
10	75.0	45.0	34	15.3	16.3	17.2	17.7	18.2	18.7	19.1	19.4	19.6	19.7	19.9
12	77.5	47.5	37	16.6	17.8	18.7	19.2	19.8	20.4	20.7	21.0	21.2	21.5	21.7
15	80.0	50.0	40	18.0	19.2	20.2	21.0	21.5	22.0	22.5	23.0	23.2	23.3	23.5
18	82.5	52.5	43	19.4	20.7	21.7	22.4	23.0	23.7	24.0	24.5	24.7	25.0	25.2
24	85.0	55.0	47	21.0	22.5	23.7	24.5	25.1	25.9	26.5	26.9	27.1	27.4	27.5
36	87.5	57.5	50	22.5	24.0	25.2	26.0	26.7	27.7	28.0	28.5	28.7	29.0	29.2
48	90.0	60.0	54	24.5	26.0	27.4	28.1	28.9	29.8	30.3	30.8	31.1	31.4	31.6

DUTY TABLES

TABLE 19.—COMPOUND NONCONDENSING STEAM ENDS

Non-jacketed, but lagged. Back pressure = 16 pounds

Boiler pressure				50.0	60.0	70.0	80.0	90.0	100.0	110.0	120.0	130.0	140.0	150.0
Absolute initial pressure				60.0	70.0	80.0	90.0	100.0	110.0	120.0	130.0	140.0	150.0	160.0
Ratio of cylinders				1.94	2.09	2.24	2.37	2.5	2.62	2.74	2.85	2.96	3.06	3.16
M.e.p. on area of high pressure cylinder				58.0	73.0	88.5	104.1	120.0	136.0	152.4	168.8	185.3	202.0	218.8
Card duty, million foot-pounds				60.0	65.0	69.5	73.5	76.5	79.5	82.0	84.0	86.0	87.5	89.0
Stroke, inches	Mechanical efficiency, per cent.	Steam efficiency, per cent.	Total efficiency, per cent.	Actual duty in million foot-pounds per 1000 pounds of steam										
6	65.0	40.0	26	15.6	16.9	18.1	19.1	19.9	20.7	21.4	21.8	22.4	22.8	23.1
8	70.0	42.5	30	18.0	19.5	20.8	22.0	22.9	23.8	24.6	25.2	25.8	26.2	26.7
10	75.0	45.0	34	20.4	22.1	23.6	25.0	26.0	27.0	27.9	28.5	29.3	29.8	30.3
12	77.5	47.5	37	22.2	24.0	25.7	27.1	28.3	29.4	30.4	31.1	31.8	32.4	33.0
15	80.0	50.0	40	24.0	26.0	27.8	29.4	30.6	31.8	32.8	33.6	34.4	35.0	35.6
18	82.5	52.5	43	25.8	28.0	29.9	31.6	32.9	34.2	35.3	36.1	37.0	37.6	38.3
24	85.0	55.0	47	28.2	30.5	32.6	34.5	36.0	37.4	38.4	39.5	40.5	41.1	41.9
36	87.5	57.5	50	30.0	32.5	34.7	36.7	38.2	39.7	41.0	42.0	43.0	43.7	44.5
48	90.0	60.0	54	32.4	35.1	37.5	39.7	41.3	42.9	44.4	45.4	46.5	47.3	48.1

DUTY TABLES

TABLE 20.—COMPOUND CONDENSING STEAM ENDS. CYLINDERS NON-JACKETED BUT LAGGED

				Back pressure = 6 pounds													
Boiler pressure				50	60	70	80	90	100	110	120	130	140	150	160	170	180
Absolute initial pressure				60	70	80	90	100	110	120	130	140	150	160	170	180	190
Ratio of cylinders				3.16	3.42	3.65	3.87	4	4	4	4	4	4	4	4	4	4
M.e.p. on area of high-pressure cylinder				82	99	116.2	133.5	151	168.5	186	203.5	221	238.5	256	273.5	291	308.5
Card duty, million foot-pounds				84	88	91	94	96.5	98	100	101.5	102.5	103.5	104.5	105.5	106.5	107
Stroke, inches	Mechanical efficiency, per cent.	Steam efficiency, per cent.	Total efficiency, per cent.	Actual duty in million foot-pounds per 1000 pounds of steam													
10	75.0	50.0	38	32.0	33.5	34.5	35.8	36.7	37.3	38	38.5	39.0	39.3	39.8	40.0	40.5	40.7
12	77.5	52.5	41	34.5	36.0	37.4	38.5	39.6	40.2	41	41.6	42.0	42.4	42.9	43.3	43.7	43.9
15	80.0	55.0	44	37.0	38.7	40.0	41.4	42.5	43.1	44	44.7	45.1	45.5	46.0	46.4	46.9	47.1
18	82.5	57.5	48	40.4	42.3	43.7	45.2	46.3	47.0	48	48.7	49.2	49.6	50.1	50.6	51.1	51.3
24	85.0	60.0	51	42.9	44.9	46.5	48.0	49.2	50.0	51	51.8	52.3	52.8	53.4	53.9	54.4	54.6
36	87.5	62.5	55	46.3	48.5	50.0	51.8	53.1	54.0	55	55.9	56.5	56.9	57.6	58.0	58.6	58.8
48	90.0	65.0	59	49.6	51.9	53.8	55.6	56.9	58.0	59	60.0	60.6	61.1	61.6	62.2	62.8	63.1

DUTY TABLES

TABLE 21.—COMPOUND CONDENSING STEAM ENDS, LOW-PRESSURE CYLINDERS JACKETED AND LAGGED

Back pressure = 6 pounds

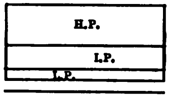
Boiler pressure.......				50	60	70	80	90	100	110	120	130	140	150	160	170	180
Absolute initial pressure...............				60	70	80	90	100	110	120	130	140	150	160	170	180	190
Ratio of cylinders....				3.16	3.42	3.65	3.87	4	4	4	4	4	4	4	4	4	4
M.e.p. on area of high-pressure cylinder				82	99	116.2	133.5	151	168.5	186	203.5	221	238.5	256	273.5	291	308.5
Card duty, million foot-pounds.........				84	88	91	94	96.5	98	100	101.5	102.5	103.5	104.5	105.5	106.5	107
Stroke, inches	Mechanical efficiency, per cent.	Steam efficiency, per cent.	Total efficiency, per cent.	Actual duty in million foot-pounds per 1000 pounds steam													
10	75.0	55.0	41	34.5	36.0	37.4	38.5	39.6	40.2	41	41.6	42.0	42.4	42.9	43.3	43.7	44.0
12	77.5	57.5	45	37.8	39.5	41.0	42.4	43.4	44.1	45	45.5	46.2	46.5	47.1	47.5	48.0	48.0
15	80.0	60.0	48	40.4	42.3	43.7	45.2	46.4	47.0	48	48.7	49.2	49.6	50.1	50.6	51.1	51.3
18	82.5	62.5	52	43.7	45.2	47.4	48.8	50.0	51.0	52	52.8	53.4	53.8	54.3	54.9	55.4	55.8
24	85.0	65.0	55	46.3	48.5	50.0	51.8	53.0	54.0	55	55.9	56.5	56.9	57.6	58.0	58.6	59.0
36	87.5	67.5	59	49.6	51.9	53.8	55.6	57.0	58.0	59	60.0	60.6	61.1	61.7	62.2	62.8	63.1
48	90.0	70.0	63	53.0	55.4	57.2	59.1	60.8	61.9	63	64.0	64.8	65.2	65.8	66.5	67.1	67.4

DUTY TABLES

TABLE 22.—TRIPLE-EXPANSION NONCONDENSING STEAM ENDS. NON-JACKETED, BUT LAGGED

Back pressure = 16 pounds

Cross-exhaust valves open

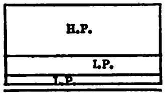


Boiler pressure				70	80	90	100	110	120	130	140	150	160	170	180	190	200
Ratio $\frac{LP}{IP}=\frac{IP}{HP}=R$				1.71	1.78	1.84	1.9	1.96	2.01	2.06	2.11	2.15	2.2	2.24	2.28	2.32	2.36
Ratio $\frac{LP}{HP}=R^2=R_1$				2.92	3.17	3.39	3.62	3.83	4.04	4.25	4.45	4.64	4.83	5.02	5.22	5.4	5.55
M.e.p. on area of high-pressure cylinder				100	118	137	156	176	196	216	236	257	278	299	320	341	363
Card duty, million ft.-lb.				78.5	83	87.5	91	94.5	97.5	100	102.5	105	107	109	111	112.5	114
Stroke, in.	Mech. eff'y, per cent.	Steam eff'y, per cent.	Total eff'y, per cent.	Actual duty in million foot-pounds per 1000 pounds steam													
10	75.0	55.0	41	32.2	34	35.9	37.4	38.7	40.0	41	42.0	43.0	44.0	44.6	45.5	46.2	46.7
12	77.5	57.5	45	35.4	37.4	39.4	41.0	42.5	44.0	45	46.2	47.2	48.0	49.0	50.0	50.6	51.3
15	80.0	60.0	48	37.7	39.9	42.0	43.7	45.4	47.0	48	49.2	50.2	51.3	52.2	53.2	54.0	54.8
18	82.5	62.5	52	40.9	43.2	45.5	47.4	49.1	50.9	52	53.4	54.8	55.8	56.8	57.8	58.7	59.2
24	85.0	65.0	55	43.2	45.6	48.1	50.0	52.0	53.8	55	56.5	57.8	59.0	60.0	61.0	62.0	62.8
36	87.5	67.5	59	46.4	49.0	51.8	53.8	55.8	57.7	59	60.6	62.0	63.1	64.2	65.6	66.5	67.1
48	90.0	70.0	63	49.5	52.2	55.0	57.2	59.6	61.6	63	64.8	66.0	67.4	68.8	70.0	71.0	71.9

DUTY TABLES

TABLE 23.—TRIPLE-EXPANSION CONDENSING STEAM ENDS. SLOW SPEED, JACKETED AND LAGGED

Back pressure = 5 pounds
No reheaters, no cutoff
Cross-exhaust valves open

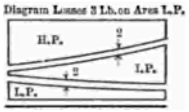


Boiler pressure				70	80	90	100	110	120	130	140	150	160	170	180	190	200
Ratio $\frac{LP}{IP} = \frac{IP}{HP} = R$				2.52	2.62	2.72	2.8	2.88	2.96	3	3	3	3	3	3	3	3
Ratio $\frac{LP}{HP} = R^2 = R_1$				6.35	6.87	7.37	7.85	8.32	8.78	9	9	9	9	9	9	9	9
M.e.p. on area of high-pressure cylinder				144.8	167.0	189.6	212.2	235.2	258.5	281.7	305	328.3	351.7	375	398.3	421.7	445
Card duty, million ft.-lb.				114	117.4	120.9	123.6	126.1	128.5	130.5	132.3	134	135.4	136.8	138	139	140.2
Stroke in.	Mech. eff'y, per cent.	Steam eff'y, per cent.	Total eff'y, per cent.	Actual duty in million foot-pounds per 1000 pounds steam													
10	75.0	67.5	50.5	57.6	59.3	61.1	62.4	63.7	64.9	65.9	66.8	67.7	68.3	69.1	69.7	70.2	70.8
12	77.5	70.0	54.0	61.6	63.4	65.3	66.7	68.1	69.4	70.5	71.4	72.4	73.1	73.9	74.5	75.1	75.7
15	80.0	72.5	58.0	66.1	68.1	70.1	71.7	73.1	74.5	75.7	76.7	77.7	78.5	79.4	80.0	80.6	81.3
18	82.5	75.0	62.0	70.7	72.8	75.0	76.6	78.2	79.7	80.9	82.0	83.1	83.9	84.8	85.6	86.2	86.9
24	85.0	77.5	66.0	75.2	77.5	79.8	81.6	83.2	84.8	86.1	87.3	88.4	89.3	90.3	91.1	91.7	92.5
36	87.5	80.0	70.0	79.8	82.2	84.6	86.5	88.3	90.0	91.3	92.6	93.8	94.7	95.8	96.6	97.3	98.1
48	90.0	82.5	74.0	84.4	86.9	89.5	91.5	93.3	95.1	96.6	97.9	99.2	100.1	101.2	102.1	102.9	103.7

DUTY TABLES

TABLE 24.—TRIPLE-EXPANSION CONDENSING STEAM ENDS. NORMAL SPEED, CROSS-EXHAUST VALVES CLOSED, JACKETED AND LAGGED

Back pressure = 4 pounds
No reheaters, no cutoff
Cross-exhaust valves closed
First intermediate space = high-pressure cylinder; second = 0.75 intermediate-pressure cylinder



Boiler Pressure				70	80	90	100	110	120	130	140	150	160	170	180	190	200
Ratio $\frac{LP}{IP} = \frac{IP}{HP} = R$				2.71	2.82	2.92	3	3	3	3	3	3	3	3	3	3	3
Ratio $\frac{LP}{HP} = R^2 = R_1$				7.37	7.97	8.55	9	9	9	9	9	9	9	9	9	9	9
M.e.p. on area of high-pressure cylinder				160	190	219	248	276	303	330	357	384	411	438	465	492	519
Card duty, million ft.-lb.				125.7	133.7	139.7	144	147.2	150	152.4	154.5	156.4	158.1	159.6	160.9	162	163
Stroke in	Mech. eff'y, per cent.	Steam eff'y, per cent.	Total eff'y, per cent.	Actual duty in million foot-pounds per 1000 pounds steam													
10	75.0	67.5	50.5	63.5	67.5	70.6	72.7	74.3	75.7	77.0	78.0	79.0	79.8	80.6	81.3	81.8	82.3
12	77.5	70.0	54.0	67.9	72.2	75.4	77.8	79.5	81.0	82.3	83.4	84.5	85.4	86.2	86.9	87.5	88.0
15	80.0	72.5	58.0	72.9	77.6	81.0	83.5	85.4	87.0	88.4	89.6	90.8	91.7	92.6	93.3	94.0	94.5
18	82.5	75.0	62.0	77.9	82.9	86.6	89.3	91.3	93.0	94.5	95.8	97.0	98.0	99.0	99.8	100.4	101.0
24	85.0	77.5	66.0	83.0	88.2	92.2	95.0	97.2	99.0	100.6	102.0	103.3	104.3	105.3	106.2	106.9	107.6
36	87.5	80.0	70.0	88.0	93.6	97.8	100.8	103.0	105.0	106.7	108.1	109.5	110.7	111.7	112.6	113.4	114.1
48	90.0	82.5	74.0	93.0	98.9	103.4	106.6	109.0	111.0	112.8	114.3	115.8	117.0	118.1	119.0	119.9	120.6

CHAPTER VIII

OPERATION AND ADJUSTMENT OF DIRECT-ACTING PUMPS

132. Setting Steam Valves on Duplex Pumps.—The general rule for setting the valves is to place the pistons, on both sides, at the center of the stroke and set the steam valves in central position with relation to the ports. As there is no lap, the outer edges of the valves will be flush with the outer edges of the ports.

In this position the valve rod clearance or the lost motion must be divided evenly, so as to have the same amount of lost motion on either side of the valve or at either end of the lost-motion link. The total lost motion may be made equal to three-tenths of the travel of the crank pin at contact stroke and can later be adjusted if the pump is not working properly.

To place the pistons at the center of the stroke, move them first to contact with one head and then with the other, each position being marked on the rod against the glands. Then place a mark midway between these two marks and move the rod until this central mark comes against the edge of the gland. This having been done on both sides, the pistons are in central position in which the valve motion arms should stand perpendicular. If this is not the case the crosshead must be disconnected and the position corrected.

Now remove the steam chest covers and place the slide valves in a position central to the ports, and adjust the lost motion so as to have the same clearance on either end. If the steam end is equipped with rotary valves a mark should be made on the hub of the crank and a corresponding one on the bearing to indicate the line-on-line position. This will save the frequent removal of the bonnets and will be a ready means of detecting any shifting. On very long pumps it is well to set the valves while the cylinders are hot, because the expansion of the steam end may throw out the setting.

133. Setting up Pumps.—In erecting a pump always be sure that it is so connected as to provide a full and uniform supply of the liquid to be handled. To accomplish this the suction pipe should be large so as to reduce the velocity, 3 to 4 feet per second being proper in a short pipe. The diameter of a long pipe should be made larger to allow for the increased friction.

The suction pipe should be as short as possible and with few bends, such as are necessary being of long radius. A suction well close to the pump and connected to the water supply by a conduit is preferable to a long suction pipe. Care must be taken to maintain a uniform grade in laying the pipe to avoid air pockets. Test the pipe to 25 pounds before covering, as small air leaks will prevent the proper working of the pump.

In all cases where a pump is to lift water by suction it will be found convenient to provide a priming pipe leading to the pump chambers; or, if the suction pipe is provided with a foot valve, this priming pipe may be led to the suction chamber (see section 32). A relief valve should then be attached to protect the foot valve. Hot water cannot be lifted by suction, as was explained in section 14.

In connecting the steam pipe make due allowance for expansion. Place a throttle valve in the pipe close to the pump and provide means for draining the pipe with the throttle valve closed. Before connecting the pump, blow out the steam pipe thoroughly to keep dirt from getting into the steam cylinders.

After stopping the pump, open all blow-off and drain valves. If it is to be idle for some time open the cock of the oil cup, so as to let the oil flow into the steam chest; then let the pump make a few quick strokes to distribute the oil well over the inside of the steam end. This will prevent rusting.

Keep the stuffing-boxes well packed with a good quality of packing. Screw up the gland no tighter than necessary to stop leakage, and renew the packing before it becomes hard. Watch the rod and outside-packed plungers to detect any cutting.

134. Operating Direct-acting Pumps.—On small pumps a pet-cock is provided on the discharge chamber for blowing out the air. On larger pumps a regular valve, the starting or waste valve, should be placed on the discharge pipe, inside of the gate and check valves. Prime and start the pump with the waste valve open, so as to work the air out of the pump chambers. If there is a check valve, this will open automatically when the right speed is reached; but if there is only a gate valve with pressure on the outside, a gage should be placed at some point inside of the gate valve. If the pump is then started, discharging through the waste valve, the gage will indicate the point at which the pressures inside and outside of the gate valve are balanced; then the gate valve may be opened.

It is essential that both sides of a duplex pump be packed exactly alike; otherwise one side will travel faster than the other.

135. Rules for Starting Large Condensing Pumps.—1. Turn on the jacket steam so as to thoroughly heat the cylinders.

2. Prime the pump bodies.

3. Start the air pump (if independent).

4. Open the throttle valve slowly. On a triple-expansion pump also open the live steam valve.

5. When the pump is under way, close the live steam valve and open the throttle valve until normal speed is obtained.

6. Adjust the cutoff valves, cushion valves and cross-exhaust valves until the pump is running smoothly at full stroke.

Do not allow a pump equipped with a surface condenser to run without water, as the exhaust steam will burn out the tube packing.

CUSHION VALVE.—Closing the cushion valves shortens the stroke. It is generally better to run with these partly open to soften the cushion.

CROSS-EXHAUST VALVES.—Opening these lengthens the stroke. In a triple-expansion engine one of the two must generally be partly or fully open in order to equalize the action of the two sides of the machine. When running very slowly it may be necessary to open both valves to maintain full stroke. Never open the intermediate cross exhaust until the high-pressure cross exhaust has been fully opened. The economy will be less with the cross-exhaust valves open and the loss will be greater with the intermediate-pressure valve open than with the high-pressure valve open.

CUTOFF ADJUSTMENT.—Triple-expansion engines are equipped with cutoff valves on the high-pressure cylinders, for the purpose of adjusting the stroke economically. If there are no guide marks on the links, note their direction of motion and that of the piston, and note whether the link is pushing the valve closed or pulling, when the piston is nearing the end of the stroke. If pushing, the cutoff is shortened by lengthening the link; if pulling, by shortening the link. This adjustment can be made while the pump is in motion. When properly adjusted the pump should make the normal stroke with equal clearances at each end.

136. Operating Worthington High-duty Pumps.—The steam valves cannot be set without instructions from the manufacturer and a diagram.

The pump is always started low duty, that is, with the cutoffs out of commission, so as to admit steam the full length of the stroke. When the engine is under way, the cutoffs are thrown in. Separate adjustment for the cutoff on each end of each cylinder is provided to control the travel at the four corners and to equalize the clearances.

The packing of the compensating cylinders and the accumulator ram needs careful attention, so as to be tight against the high pressure without undue friction. Moreover, the air piston of the accumulator should be kept flooded with oil to prevent excessive air leakage.

Open the petcocks on the compensating cylinders occasionally to let the air out and prevent an explosion should the pump make a quick stroke. Keep the air chamber well supplied with air and the accumulator piston in a central position.

APPENDIX

TABLE 1.—FRICTION OF PENNSYLVANIA OIL, 38° BAUMÉ

Pounds per Square Inch in Pipe—10 English Miles Long

$d =$	2	3	4	5	6	8	10
$d^5 =$	32	243	1,024	3,125	7,776	32,768	10,000
$d^{2.5} =$	5.66	15.6	32	55.9	88.2	181	316.2
Barrels per hour, B	P						
10	28	3.7	0.88	0.288	0.116	0.0274	0.0091
15	62	8.3	1.98	0.65	0.26	0.0641	0.0203
20	112	14.8	3.52	1.15	0.462	0.11	0.036
25	176	23.0	5.5	1.8	0.725	0.172	0.056
30	253	33.2	7.92	2.6	1.04	0.247	0.081
35	345	45.2	10.8	3.53	1.42	0.337	0.112
40	450	59.0	14.3	4.6	1.85	0.44	0.144
45	570	74.9	17.8	5.8	2.35	0.556	0.181
50	700	92.4	22.0	7.2	2.9	0.686	0.225
55	850	112.0	22.6	8.7	3.5	0.84	0.273
60	1010	133.0	31.6	10.4	4.16	0.99	0.325
65	1184	156.0	37.2	12.2	4.9	1.16	0.381
70	1380	181.0	43.0	14.1	5.68	1.36	0.442
75	1580	208.0	49.5	16.2	6.51	1.55	0.508
80	1800	236.0	56.2	18.4	7.4	1.76	0.578
85	2030	267.0	63.5	20.8	8.36	1.98	0.652
90	2270	300.0	71.1	23.3	9.39	2.24	0.73
95	2530	334.0	79.2	26.0	10.45	2.48	0.814
100	2800	370.0	88.0	28.8	11.6	2.75	0.902
125	4400	578.0	137.0	45.0	18.1	4.3	1.42
150	6300	830.0	198.0	65.0	26.0	6.18	2.03
175	....	1130.0	270.0	88.0	35.4	8.4	2.75
200	....	1480.0	352.0	115.0	46.3	11.0	3.6
225	....	1870.0	425.0	146.0	58.5	13.9	4.55
250	....	2300.0	550.0	180.0	72.2	17.2	5.6
275	....	2800.0	665.0	217.0	87.5	20.8	6.8
300	....	3320.0	792.0	260.0	104.0	24.7	8.1
325	....	3900.0	930.0	304.0	122.0	29.0	9.5
350	....	4530.0	1020.0	353.0	142.0	33.7	11.2
375	....	....	1240.0	405.0	163.0	38.5	12.6
400	....	....	1410.0	460.0	185.0	44.0	14.4
425	....	....	1590.0	520.0	209.0	49.6	16.3
450	....	....	1780.0	582.0	235.0	55.6	18.3
475	....	....	1980.0	650.0	261.0	62.5	20.4
500	....	....	2200.0	720.0	289.0	68.8	22.5
525	....	....	2420.0	795.0	319.0	75.8	24.8
550	....	....	2660.0	870.0	350.0	83.0	27.3

TABLE 1.—FRICTION OF PENNSYLVANIA OIL, 38° BAUMÉ—*Continued*

Pounds per Square Inch in Pipe—10 English Miles Long

$d =$	5	6	8	10	$d =$	8	10
$d^5 =$	3,125	7,776	32,768	10,000	$d^5 =$	32,768	10,000
$d^{2.5} =$	55.9	88.2	181	316.2	$d^{2.5} =$	181	316.2
Barrels per hour, B	P				Barrels per hour, B		
550	870	350	83	27.2	3600	3560	1160
575	950	384	91	29.8	3800	3970	1300
600	1040	417	99	32.4	4000	4400	1440
625	1120	454	107	35.0	4200	....	1580
650	1220	490	116	38.0	4400	....	1740
675	1310	530	125	41.0	4600	....	1920
700	1410	569	135	44.2	4800	....	2070
725	1520	610	144	47.3	5000	....	2250
750	1620	653	155	50.6	5200	....	2430
775	1730	698	165	54.1	5400	....	2620
800	1840	743	176	57.6	5600	....	2820
825	1960	790	187	61.2	5800	....	3030
850	2080	839	198	65.0	6000	....	3240
875	2210	888	211	68.9	6200	....	3460
900	2330	940	223	73.0	6400	....	3680
925	2470	992	235	77.0	6600	....	3920
950	2600	1045	248	81.2	6800		4160
975	2740	1110	262	85.5	7000	....	4410
1000	2880	1158	275	90.0	....	....	....
1100	3480	1400	332	109.0	....	....	....
1200	4150	1665	395	130.0	....	....	....
1300	4860	1955	464	152.0	....	....	....
1400	....	2270	538	176.0	....	....	....
1500	....	2610	618	203.0	....	....	....
1600	....	2960	704	231.0	....	....	....
1700	....	3350	793	261.0	....	....	....
1800	....	3750	890	292.0	....	....	....
1900	....	4180	990	325.0	....	....	....
2000	....	4630	1100	361.0	....	....	....
2200	....	....	1330	436.0	....	....	....
2400	....	....	1580	520.0	....	....	....
2600	....	....	1860	610.0	....	....	....
2800			2160	706.0			
3000	....	....	2475	811.0	....	....	....
3200	....	....	2840	922.0	....	....	....
3400	....	....	3180	1040.0	....	....	....
3600	....	....	3560	1170.0	....	....	....

Table 2.—Friction of Heavy California Oil in 4-Inch Pipe

p = Pounds per Square Inch in 1 English Mile—$8^5 \div 4^5 = 32$

Barrels per hour, B	Temperature, deg. F. = t					
	50	60	70	80	90	100
5	222	55.4	24.6	13.9	8.87	6.16
10	558	140.0	53.1	34.9	22.4	15.5
15	959	240.0	107.0	60.0	38.4	26.7
20	1,410	352.0	156.0	88.0	56.3	39.1
25	1,900	474.0	211.0	119.0	75.9	52.8
30	2,416	602.0	269.0	151.0	96.7	67.2
35	2,968	742.0	330.0	186.0	119.0	82.5
40	3,546	887.0	394.0	222.0	142.0	98.5
45	4,152	1,038.0	461.0	260.0	166.0	115.0
50	4,768	1,194.0	531.0	299.0	191.0	133.0
60	6,090	1,523.0	677.0	381.0	244.0	169.0
70	7,480	1,870.0	831.0	467.0	299.0	208.0
80	8,934	2,234.0	993.0	559.0	358.0	248.0
90	10,460	2,615.0	1,162.0	654.0	419.0	291.0
100	12,030	3,010.0	1,338.0	752.0	483.0	336.0
150	20,670	5,184.0	2,298.0	1,293.0	824.0	576.0
200	30,340	7,584.0	3,360.0	1,898.0	1,213.0	842.0
250	41,030	10,270.0	4,577.0	2,567.0	1,642.0	1,140.0
300	52,070	13,030.0	5,790.0	3,264.0	2,083.0	1,447.0
350	63,940	16,000.0	7,110.0	4,000.0	2,567.0	1,778.0
400	76,400	19,100.0	8,480.0	4,770.0	3,060.0	2,120.0
450	89,400	22,400.0	9,950.0	5,600.0	3,590.0	2,480.0
500	103,000	25,700.0	11,400.0	6,430.0	4,130.0	2,730.0
550	....	29,200.0	12,980.0	7,300.0	4,670.0	3,245.0
600	....	32,740.0	14,600.0	8,190.0	5,250.0	3,650.0
650	....	36,500.0	16,220.0	9,130.0	5,840.0	4,055.0
700	....	40,300.0	17,900.0	10,080.0	6,470.0	4,480.0
750	....	44,160.0	19,630.0	11,040.0	7,070.0	4,910.0
800	....	48,100.0	21,400.0	12,030.0	7,710.0	5,340.0
850	....	52,260.0	23,300.0	13,060.0	8,360.0	5,810.0
900	....	56,300.0	25,000.0	14,100.0	9,030.0	6,270.0
1,000	....	64,800.0	28,800.0	16,200.0	10,400.0	7,200.0

TABLE 2.—FRICTION OF HEAVY CALIFORNIA OIL IN 4-INCH PIPE
(*Continued*)

p = Pounds per Square Inch in 1 English Mile—$8^5 \div 4^5 = 32$

Barrels per hour, B	Temperature, deg. F. = t					
	110	120	130	140	150	160
5	4.53	3.35	2.74	2.22	1.85	1.54
10	11.4	8.43	6.9	5.58	4.79	3.88
15	19.6	14.5	11.9	9.59	8.01	6.67
20	28.8	21.2	17.4	14.1	11.8	9.77
25	38.7	28.6	23.4	19.0	15.8	13.2
30	49.4	36.5	29.9	24.2	20.2	16.8
35	60.6	44.8	36.7	29.7	24.8	20.6
40	72.5	53.5	43.8	35.5	29.6	24.6
45	84.8	62.6	51.3	41.5	34.6	28.8
50	97.6	72.0	58.9	47.7	40.0	33.3
60	125.0	91.9	75.2	60.9	50.8	42.3
70	153.0	113.0	92.4	74.8	62.4	52.0
80	183.0	135.0	110.0	89.4	74.6	62.1
90	214.0	158.0	129.0	105.0	87.3	72.6
100	246.0	182.0	149.0	120.0	101.0	83.6
150	423.0	312.0	255.0	207.0	173.0	144.0
200	621.0	458.0	375.0	304.0	253.0	211.0
250	839.0	621.0	506.0	410.0	343.0	285.0
300	1050.0	787.0	643.0	522.0	435.0	362.0
350	1310.0	967.0	791.0	640.0	534.0	445.0
400	1560.0	1150.0	944.0	765.0	637.0	531.0
450	1830.0	1390.0	1110.0	896.0	746.0	621.0
500	2100.0	1550.0	1270.0	1030.0	858.0	714.0
550	2390.0	1760.0	1442.0	1168.0	975.0	811.0
600	2680.0	1980.0	1620.0	1310.0	1100.0	909.0
650	2980.0	2210.0	1800.0	1460.0	1220.0	1020.0
700	3300.0	2430.0	1990.0	1610.0	1340.0	1120.0
750	3610.0	2670.0	2180.0	1770.0	1480.0	1230.0
800	3940.0	2910.0	2380.0	1930.0	1610.0	1340.0
850	4270.0	3150.0	2580.0	2090.0	1750.0	1460.0
900	4610.0	3390.0	2780.0	2250.0	1880.0	1570.0
1000	5310.0	3910.0	3200.0	2590.0	2170.0	1800.0

TABLE 3.—FRICTION OF HEAVY CALIFORNIA OIL IN 6-INCH PIPE

p = Pounds per Square Inch in 1 English Mile—$8^5 \div 6^5 = 4.214$

Barrels per hour, B	Temperature, deg. F. = t					
	50	60	70	80	90	100
10	73.6	18.4	7.0	4.6	3.0	2.05
15	126.0	31.6	14.0	7.9	5.06	3.51
20	186.0	46.3	18.6	11.6	7.42	5.15
25	250.0	62.4	27.7	15.6	10.0	6.96
30	318.0	79.6	35.4	20.0	12.7	8.84
35	391.0	99.7	43.5	24.4	15.7	10.9
40	467.0	117.0	51.9	29.2	18.7	13.0
45	547.0	137.0	60.7	34.2	21.9	15.2
50	628.0	157.0	70.0	39.3	25.2	17.5
60	802.0	201.0	89.1	50.2	32.1	22.3
70	985.0	247.0	110.0	61.6	39.4	27.4
80	1,177.0	294.0	131.0	73.6	47.1	32.7
90	1,377.0	344.0	153.0	86.0	55.1	38.0
100	1,585.0	396.0	176.0	99.1	63.6	44.3
150	2,723.0	683.0	303.0	170.0	109.0	75.9
200	3,995.0	999.0	443.0	250.0	160.0	111.0
250	5,404.0	1,353.0	603.0	338.0	216.0	150.0
300	6,857.0	1,715.0	763.0	430.0	275.0	191.0
350	8,420.0	2,107.0	936.0	527.0	338.0	234.0
400	10,060.0	2,516.0	1,117.0	628.0	403.0	280.0
450	11,780.0	2,946.0	1,311.0	738.0	472.0	327.0
500	13,550.0	3,388.0	1,505.0	847.0	544.0	377.0
600	17,250.0	4,310.0	1,918.0	1,079.0	691.0	481.0
700	21,220.0	5,306.0	2,360.0	1,328.0	851.0	590.0
800	25,350.0	6.338.0	2,820.0	1,585.0	1,016.0	704.0
900	29,660.0	7,417.0	3,296.0	1,854.0	1,188.0	826.0
1,000	34,140.0	8,534.0	3,793.0	2,133.0	1,366.0	948.0
1,200	43,530.0	10,880.0	4,838.0	2,723.0	1,740.0	1,210.0
1,400	53,440.0	13,360.0	5,942.0	3,342.0	2,141.0	1,488.0
1,600	63,900.0	15,970.0	7,098.0	3,992.0	2,554.0	1,774.0
1,800	74,760.0	18,690.0	8,303.0	4,678.0	3,000.0	2,078.0
2,000	86,000.0	21,510.0	9,560.0	5,378.0	3,443.0	2,390.0

TABLE 3.—FRICTION OF HEAVY CALIFORNIA OIL IN 6-INCH PIPE
(*Continued*)

p = Pounds per Square Inch in 1 English Mile—$8^5 \div 6^5 = 4.214$

Barrels per hour, B	Temperature, deg. F. = t					
	110	120	130	140	150	160
10	1.5	1.11	0.908	0.735	0.614	0.511
15	2.58	1.91	1.56	1.26	1.06	0.878
20	3.79	2.8	2.29	1.85	1.55	1.29
25	5.1	3.78	3.08	2.5	2.08	1.73
30	6.5	4.8	3.93	3.18	2.66	2.21
35	7.98	5.9	4.83	3.91	3.26	2.72
40	9.54	7.05	5.77	4.67	3.9	3.24
45	11.2	8.25	6.75	5.47	4.56	3.8
50	12.9	9.48	7.76	6.28	5.27	4.38
60	16.4	12.1	9.9	8.02	6.69	5.57
70	20.1	14.9	12.2	9.85	8.22	6.84
80	24.0	17.8	14.5	11.8	9.82	8.17
90	28.1	20.8	17.0	13.8	11.5	9.56
100	32.4	24.0	19.6	15.9	13.2	11.0
150	55.7	41.1	33.6	27.2	22.7	18.9
200	81.8	60.3	49.3	40.0	33.4	27.7
250	111.0	81.8	66.6	54.0	45.1	37.6
300	138.0	104.0	84.7	68.7	57.3	47.6
350	172.0	127.0	104.0	84.3	70.4	58.6
400	206.0	152.0	124.0	101.0	83.9	70.0
450	241.0	178.0	147.0	118.0	98.2	81.8
500	277.0	205.0	167.0	136.0	113.0	94.0
600	353.0	261.0	213.0	173.0	144.0	120.0
700	434.0	320.0	262.0	213.0	177.0	148.0
800	519.0	383.0	313.0	254.0	212.0	176.0
900	607.0	447.0	366.0	297.0	248.0	206.0
1000	700.0	514.0	422.0	342.0	285.0	237.0
1200	889.0	658.0	540.0	434.0	363.0	302.0
1400	1090.0	809.0	662.0	535.0	447.0	371.0
1600	1310.0	965.0	788.0	641.0	535.0	443.0
1800	1525.0	1130.0	923.0	750.0	624.0	519.0
2000	1760.0	1300.0	1060.0	860.0	721.0	599.0

TABLE 4.—FRICTION OF HEAVY CALIFORNIA OIL IN 8-INCH PIPE

p = Pounds per Square Inch in 1 English Mile

Barrels per hour, B	Temperature, deg. F. = t					
	50	60	70	80	90	100
25	59.2	14.8	6.58	3.7	2.37	1.65
50	149.0	37.3	16.6	9.33	5.97	4.15
75	256.0	64.1	28.5	16.0	10.3	7.12
100	376.0	94.0	41.8	23.5	15.1	10.5
150	646.0	162.0	71.8	40.4	25.8	18.0
200	948.0	237.0	105.0	59.3	37.9	26.3
250	1,282.0	321.0	143.0	80.2	51.3	35.6
300	1,627.0	407.0	181.0	102.0	65.1	45.2
350	1,998.0	500.0	222.0	125.0	80.2	55.5
400	2,387.0	597.0	265.0	149.0	95.5	66.3
450	2,794.0	699.0	311.0	175.0	112.0	77.6
500	3,215.0	804.0	357.0	201.0	129.0	89.3
600	4,093.0	1,023.0	455.0	256.0	164.0	114.0
700	5,034.0	1,259.0	560.0	315.0	202.0	140.0
800	6,015.0	1,504.0	669.0	376.0	241.0	167.0
900	7,038.0	1,760.0	782.0	440.0	282.0	196.0
1,000	8,100.0	2,025.0	900.0	506.0	324.0	225.0
1,100	9,198.0	2,300.0	1,022.0	575.0	368.0	256.0
1,200	10,330.0	2,582.0	1,148.0	646.0	413.0	287.0
1,300	11,500.0	2,874.0	1,277.0	719.0	460.0	319.0
1,400	12,680.0	3,171.0	1,410.0	793.0	508.0	353.0
1,500	13,910.0	3,477.0	1,545.0	869.0	557.0	387.0
1,600	15,160.0	3,789.0	1,684.0	947.0	606.0	421.0
1,700	16,430.0	4,109.0	1,826.0	1,027.0	658.0	457.0
1,800	17,740.0	4,435.0	1,971.0	1,110.0	712.0	493.0
1,900	19,060.0	4,765.0	2,118.0	1,191.0	763.0	530.0
2,000	20,410.0	5,103.0	2,268.0	1,276.0	817.0	567.0
2,200	23,180.0	5,794.0	2,575.0	1,449.0	927.0	644.0
2,400	26,030.0	6,506.0	2,892.0	1,627.0	1,041.0	723.0
2,600	28,960.0	7,240.0	3,218.0	1,810.0	1,158.0	805.0
2,800	31,970.0	7,994.0	3,552.0	1,998.0	1,279.0	888.0
3,000	35,050.0	8,764.0	3,895.0	2,191.0	1,406.0	974.0

TABLE 4.—FRICTION OF HEAVY CALIFORNIA OIL IN 8-INCH PIPE (*Continued*)

p = Pounds per Square Inch in 1 English Mile

Barrels per hour, B	Temperature, deg. F. = t					
	110	120	130	140	150	160
25	1.21	0.894	0.731	0.592	0.494	0.411
50	3.05	2.25	1.84	1.49	1.25	1.04
75	5.23	3.87	3.16	2.56	2.14	1.78
100	7.68	5.68	4.64	3.76	3.14	2.61
150	13.2	9.74	7.97	6.46	5.39	4.49
200	19.4	14.3	11.7	9.48	7.91	6.58
250	26.2	19.4	15.8	12.8	10.7	8.91
300	32.6	24.6	20.1	16.3	13.6	11.3
350	40.8	30.2	24.7	20.0	16.7	13.9
400	48.8	36.0	29.5	23.9	19.9	16.6
450	57.1	42.2	34.5	28.0	23.3	19.4
500	65.7	48.5	39.7	32.2	26.8	22.3
600	83.6	61.8	50.5	40.9	34.2	28.4
700	103.0	76.0	62.2	50.4	42.0	35.0
800	123.0	90.8	74.3	60.2	50.2	41.8
900	144.0	106.0	86.9	70.4	58.8	48.9
1,000	166.0	122.0	100.0	81.0	67.6	56.3
1,100	188.0	139.0	114.0	92.0	76.8	63.9
1,200	211.0	156.0	128.0	103.0	86.2	71.7
1,300	235.0	174.0	141.0	115.0	96.0	79.8
1,400	259.0	192.0	157.0	127.0	106.0	88.1
1,500	284.0	210.0	172.0	139.0	116.0	96.6
1,600	310.0	229.0	187.0	152.0	127.0	105.0
1,700	336.0	248.0	203.0	165.0	137.0	114.0
1,800	362.0	268.0	219.0	178.0	148.0	123.0
1,900	389.0	288.0	235.0	191.0	159.0	133.0
2,000	417.0	308.0	252.0	204.0	171.0	142.0
2,200	473.0	350.0	286.0	232.0	194.0	161.0
2,400	532.0	393.0	321.0	260.0	217.0	181.0
2,600	592.0	437.0	358.0	290.0	242.0	201.0
2,800	653.0	482.0	395.0	320.0	267.0	222.0
3,000	716.0	529.0	433.0	351.0	293.0	244.0

TABLE 5.—FRICTION OF HEAVY CALIFORNIA OIL IN 10-INCH PIPE

p = Pounds per Square Inch in 1 English Mile—$8^5 \div 10^5 = 0.32768$

Barrels per hour, B	Temperature, deg. F. = t					
	50	60	70	80	90	100
50	48.8	12.2	5.44	3.06	1.96	1.36
75	83.9	21.0	9.34	5.25	3.38	2.33
100	123.0	30.8	13.7	7.7	4.95	3.44
150	212.0	53.1	23.5	13.3	8.46	5.9
200	311.0	77.7	34.4	19.5	12.4	8.62
250	420.0	105.0	46.9	26.3	16.8	11.7
300	533.0	134.0	59.3	33.4	21.4	14.8
350	655.0	164.0	72.8	41.0	26.3	18.2
400	782.0	196.0	86.9	48.9	31.3	21.7
450	916.0	229.0	102.0	57.4	36.7	25.4
500	1,055.0	264.0	117.0	65.9	42.3	29.3
600	1,342.0	335.0	149.0	83.9	53.8	37.4
700	1,650.0	413.0	184.0	103.0	66.2	45.9
800	1,971.0	493.0	219.0	123.0	79.0	54.4
900	2,305.0	577.0	256.0	144.0	92.4	64.2
1,000	2,655.0	664.0	295.0	166.0	106.0	73.8
1,100	3,014.0	754.0	335.0	189.0	121.0	83.9
1,200	3,385.0	846.0	376.0	212.0	135.0	94.0
1,300	3,770.0	942.0	419.0	236.0	151.0	105.0
1,400	4,156.0	1,040.0	462.0	260.0	167.0	116.0
1,500	4,560.0	1,140.0	506.0	285.0	183.0	127.0
1,600	4,968.0	1,242.0	552.0	310.0	199.0	138.0
1,700	5,385.0	1,347.0	599.0	337.0	216.0	140.0
1,800	5,814.0	1,454.0	646.0	364.0	234.0	162.0
1,900	6,246.0	1,562.0	694.0	390.0	250.0	174.0
2,000	6,688.0	1,672.0	743.0	418.0	268.0	186.0
2,500	9,054.0	2,264.0	1,006.0	566.0	362.0	252.0
3,000	11,490.0	2,872.0	1,276.0	718.0	461.0	319.0
3,500	14,100.0	3,526.0	1,567.0	882.0	564.0	392.0
4,000	16,860.0	4,215.0	1,873.0	1,054.0	674.0	468.0
4,500	19,720.0	4,932.0	2,190.0	1,233.0	789.0	548.0
5,000	22,700.0	5,671.0	2,522.0	1,418.0	908.0	631.0

TABLE 5.—FRICTION OF HEAVY CALIFORNIA OIL IN 10-INCH PIPE
(*Continued*)

p = Pounds per Square Inch in 1 English Mile—$8^5 + 10^5 = 0.32768$

Barrels per hour, *B*	Temperature, deg. F. = *t*					
	110	120	130	140	150	160
50	1.0	0.737	0.603	0.488	0.41	0.341
75	1.72	1.27	1.04	0.839	0.701	0.583
100	2.52	1.86	1.52	1.23	1.03	0.855
150	4.33	3.19	2.61	2.12	1.77	1.47
200	6.36	4.69	3.84	3.11	2.59	2.16
250	8.59	6.36	5.18	4.2	3.51	2.92
300	10.7	8.06	6.59	5.34	4.46	3.7
350	13.4	9.9	8.1	6.56	5.47	4.56
400	16.0	11.8	9.67	7.83	6.52	5.44
450	18.7	13.8	11.3	9.18	7.64	6.36
500	21.5	15.9	13.0	10.6	8.78	7.31
600	27.4	20.3	16.6	13.4	11.2	9.31
700	33.8	24.9	20.4	16.5	13.8	11.5
800	40.3	29.8	24.4	19.7	16.5	13.7
900	47.2	34.8	28.5	23.1	19.3	16.2
1000	54.4	40.0	32.8	26.6	22.2	18.5
1100	61.6	45.6	37.4	30.2	25.2	21.0
1200	69.2	51.1	42.0	33.8	28.3	23.5
1300	77.0	57.0	46.2	37.7	31.5	26.2
1400	84.9	62.9	51.5	41.6	34.8	28.9
1500	93.1	68.8	56.4	45.6	38.0	31.7
1600	102.0	75.1	61.3	49.8	41.6	34.4
1700	110.0	81.3	66.6	54.1	44.9	37.4
1800	119.0	87.8	71.8	58.3	48.5	40.3
1900	128.0	94.4	77.0	62.6	52.1	43.6
2000	137.0	101.0	82.6	66.9	56.1	46.6
2500	185.0	137.0	112.0	90.6	75.6	62.9
3000	235.0	174.0	142.0	115.0	96.0	80.0
3500	288.0	213.0	174.0	141.0	118.0	98.0
4000	344.0	254.0	208.0	169.0	141.0	117.0
4500	403.0	298.0	244.0	197.0	165.0	137.0
5000	464.0	342.0	280.0	227.0	190.0	158.0

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Abbreviations: p.—pump; d.a.—direct-acting.

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